STEAM ENGINES AND AIR COMPRESSORS

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Steam Engines and Air Compressors

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STEAM ENGINES

COMPRESSED AIR IN COAL MINING

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NOTE.—This book is made up of separate parts, or sections, as indicated by their titles, and the page numbers of each usually begin with 1. In this list of contents the titles of the parts are given in the order in which they appear in the book, and under each title is a full synopsis of the subjects treated.

COMPRESSED AIR IN COAL MINING

STEAM ENGINES

Serial 46 —————— Edition 5

GENERAL MINING TYPES

INTRODUCTION

1. Definition.—An **engine** is a machine for converting energy into mechanical power. A **steam engine** is one in which the motive power is steam.

2. Requirements of an Engine.—The conditions under which a steam engine is used about a mine are usually unfavorable. Dust and dirt are more or less abundant, and if the journals and bearings of a mine engine are not carefully watched, they may heat and wear quickly. This is important, for if one of the principal hoisting, haulage, fan, or pumping engines should suddenly break down, it might cause a stoppage of the entire plant, particularly since few mines are equipped with shops sufficiently complete to quickly repair large engines. Hence, the essential points to be considered in the choice of an engine for mine work are simplicity of construction, durability, accessibility to its parts, and in case of necessity, the ease with which worn and broken parts can be duplicated or repaired.

3. Classification of Steam Engines.—Steam engines are classified in several different ways, and often a full description of an engine requires that reference be made to several of these classes or groups. Following are the most important classifications of engines:

(*a*) According as the engine is movable or fixed	Portable Stationary

(*b*)	According to the kind of service required	Hoisting Haulage Pumping Fan Locomotive, etc.
(*c*)	According to the number of engines on one shaft	Single Duplex
(*d*)	According to the use of steam and arrangement of cylinders	Simple Compound Triple-expansion Quadruple-expansion
(*e*)	According to manner of exhaust	Condensing Non-condensing
(*f*)	According to the type of valve used, as	Plain slide valve Adjustable cut-off Automatic cut-off
(*g*)	According to motion of piston	Reciprocating Rotary

4. Types of Engines.—Engines may be horizontal, vertical, or inclined; direct-acting (first-motion), or geared (second-motion); single-acting, or double-acting. All these different types involve essentially the same principles, and therefore a description of a few of them will be sufficient to give a general knowledge of the principles they involve.

5. Principle of the Steam Engine.—The action of a steam engine depends on the utilization of the energy of steam under pressure. Steam is alternately admitted to, and exhausted from, the two ends of a cylinder containing the piston. The piston and the piston rod are thus given a to-and-fro, or reciprocating, motion that is converted into rotary motion by the crank and connecting-rod.

6. Parts of a Simple Engine.—A simple steam engine consists of the following fundamental parts: (1) A steam cylinder and steam chest with necessary ports and valves; (2) a piston and piston rod with crosshead and guides; (3) a connecting-rod; (4) a crank and crank-shaft, with its bearing and flywheel; (5) an eccentric and rods, or some other

device for operating the steam and exhaust valves. The following description of a simple slide-valve engine will make clear the action and relationship of the parts. The slide-valve engine is of comparatively simple construction and may be considered as a type of all reciprocating steam engines.

SIMPLE SLIDE-VALVE ENGINE

7. General Principles.—Fig. 1 (*a*) shows a diagrammatic plan of a simple slide-valve engine, the cylinder or

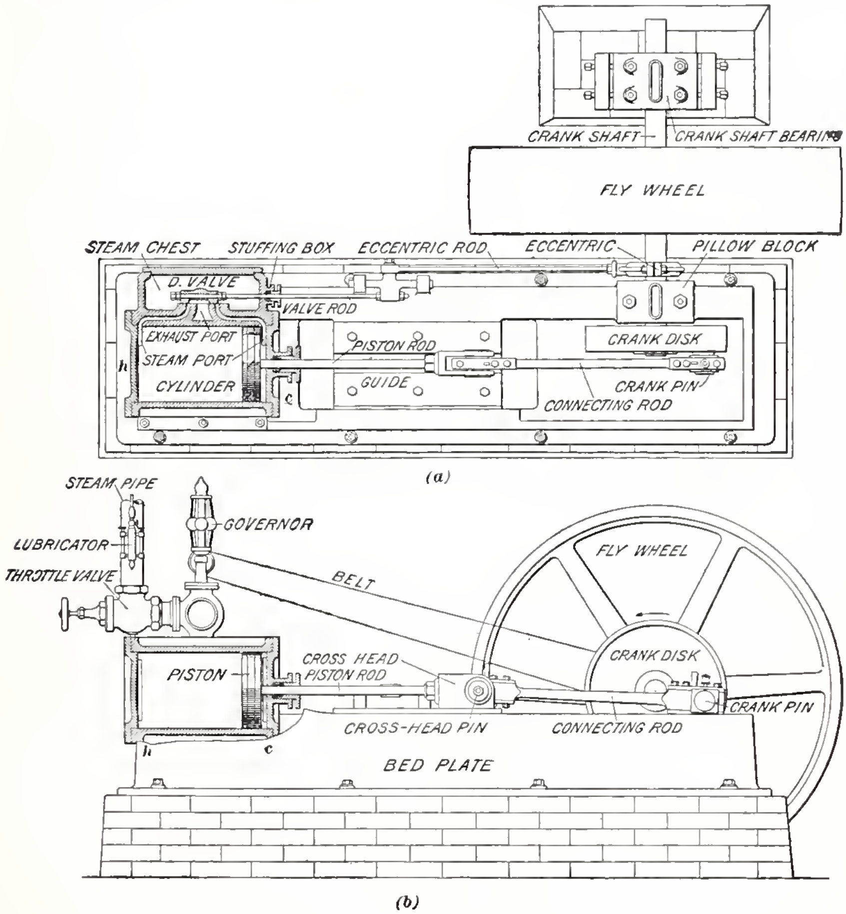


FIG. 1

steam chest, and slide or **D** valve in section ; Fig. 1 (*b*) is a diagrammatic elevation of the same, the various parts being

named on the figure. Steam from the boiler enters the steam chest through the throttle valve. The **D** valve is in such a position that steam must enter the crank end *c* of the cylinder, and since the head end *h* is open to the atmosphere, the steam pressure forces the piston from *c* toward *h*. If the steam in the end *c* be allowed to escape into the atmosphere, and steam be admitted to the end *h*, it will force the piston back again from *h* to *c*, and so on. The action of the valve that controls the admission and exhaust of steam from the cylinder will be explained in detail farther on.

8. Expansion of Steam in Cylinder.—The valve controlling the entry of steam into the cylinder can be so arranged as to permit steam from the boiler to enter the cylinder during the full stroke of the piston, but ordinarily the valve is so arranged that steam is admitted during a portion of the stroke only, the valve then closing and shutting off the supply. This point of the stroke is called the **point of cut-off.** The live steam in the cylinder when the engine is under full load is at nearly boiler pressure when the cut-off takes place, after which the steam expands, and by so doing continues to move the piston forwards but at a decreasing pressure. By thus using the steam expansively, it is evident that a smaller amount is required for each stroke of the piston than would be necessary if steam were admitted during the full stroke.

9. Change of Reciprocating Into Rotary Motion. The piston rod, Fig. 1 (*a*) and (*b*), connects with the piston at one end and with the crosshead at the other. The crosshead slides between guides, and keeps the piston rod in line with the cylinder. The connecting-rod is attached at one end to the crosshead by a wristpin, or crosshead pin, and at the other end to the crankpin. The crank is keyed to the crank-shaft, which rests in shaft bearings, or pillow-blocks. It is evident that the end of the piston rod can move only in a straight line, and, since the crank-shaft must rotate in its bearings, that the crank and crankpin can move only in a circle. It will thus be seen that by means of the

several parts shown, the to-and-fro, or reciprocating, motion of the piston is changed into the circular motion of the crank and the crank-shaft. The crank-shaft may be coupled directly to the machinery to be driven, or a pulley or gear may be fastened to the shaft and power thus transmitted.

When an engine runs in the direction indicated by the arrow in the figure, it is said to *run under;* when it runs in the opposite direction, it is said to *run over.*

PRINCIPAL PARTS OF A SLIDE-VALVE ENGINE

10. Fig. 2 shows a perspective of a common type of

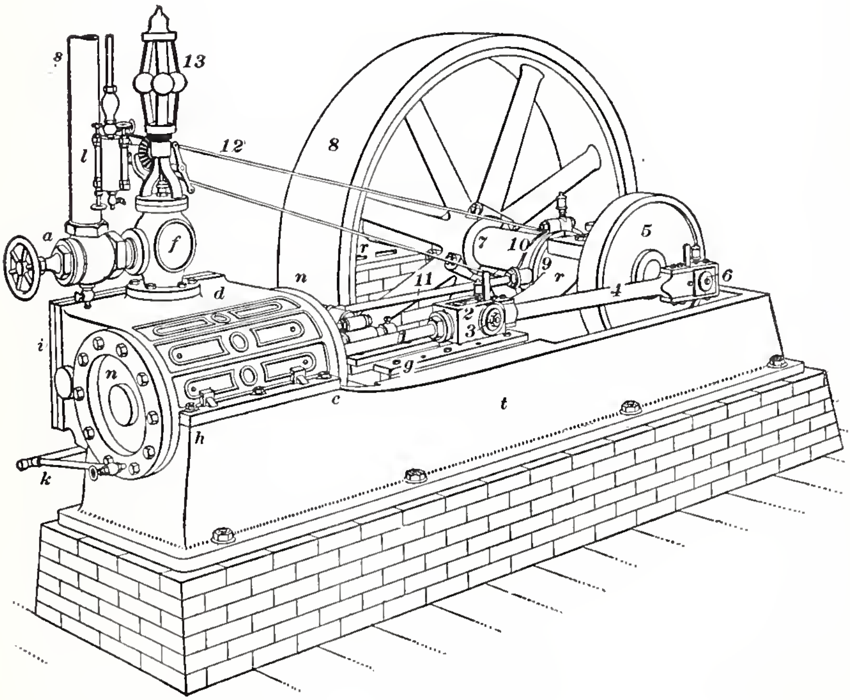


Fig. 2

simple slide-valve engine. The principal parts are indicated as follows:

STATIONARY PARTS		MOVABLE PARTS	
a,	Throttle valve	*1*,	Piston rod
c,	Crank end of cylinder	*2*,	Crosshead
d,	Steam chest	*3*,	Crosshead pin
f,	Governor	*4*,	Connecting-rod
g,	Guide bar	*5*,	Crank
h,	Head end of cylinder	*6*,	Crankpin
i,	Steam-chest cap	*7*,	Crank-shaft
k,	Drain pipe	*8*,	Flywheel
l,	Cylinder lubricator	*9*,	Eccentric
n,	Cylinder heads	*10*,	Eccentric strap
r,	Shaft bearings	*11*,	Eccentric rod
s,	Steam pipe	*12*,	Governor belt
t,	Bed, or frame	*13*,	Governor balls

11. Steam Cylinder.—The **steam cylinder** of an engine, Fig. 3, is a hollow cast-iron cylinder provided with

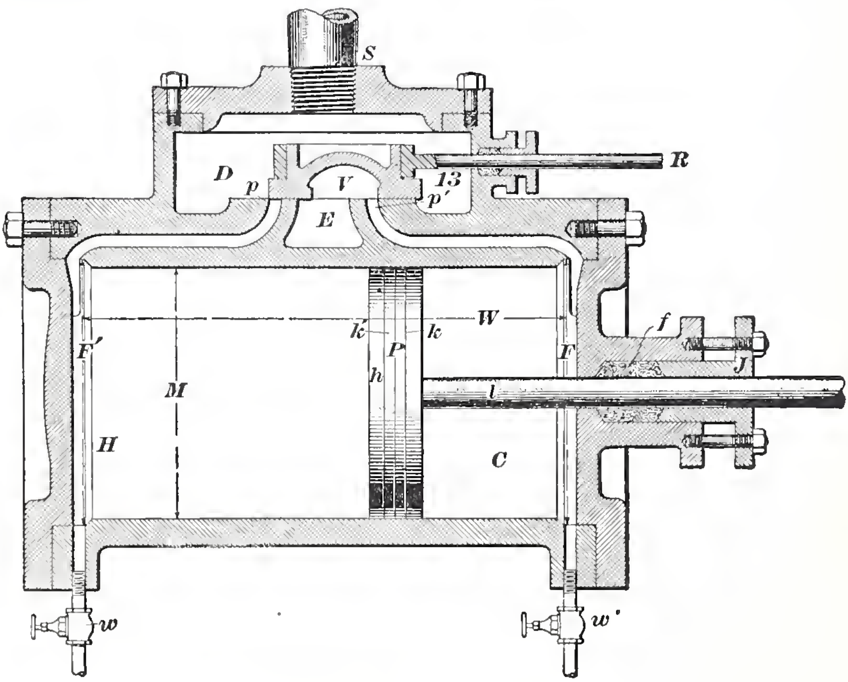


FIG. 3

openings *p* and *p'* at each end called ports. These ports are opened alternately to the boiler and to the atmosphere or to

a condenser. When the port p' is open to the atmosphere the exhaust steam filling the crank end of the cylinder escapes into the atmosphere. When the port p is open to the boiler, the live steam enters the head end of the cylinder from the boiler. The two ends of the cylinder are counterbored for a short distance, that is, the bore, or diameter, FF' is made greater than the usual diameter M of the cylinder. The piston projects partly over this counterbore at the end of each stroke, the object being to prevent the formation of a shoulder on the smooth inside surface of the cylinder at the end of the stroke. Fig. 4 shows the counterbore and the shape of the steam ports.

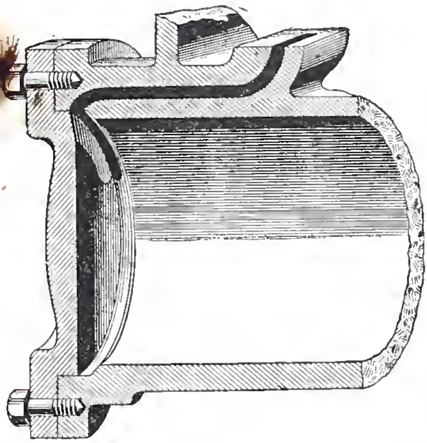


FIG. 4

The **stroke** of an engine is the distance passed over by the piston in passing from one end of the cylinder to the other, and is equal to the diameter of the circle described by the center of the crankpin. The stroke from the head to the crank end of the cylinder is the **forward stroke,** and from the crank end to the head end the **return stroke.**

The size of an engine is generally expressed by giving its diameter of cylinder and stroke in inches; thus, an engine having a cylinder diameter of 16 inches and a stroke of 24 inches, is called a $16'' \times 24''$ engine. It is customary to give the diameter first.

12. Clearance. — The distance between the cylinder heads is slightly greater than the working length, since a small space must be left between the piston and each head at the end of either stroke to avoid the possibility of the piston knocking out a cylinder head. This space, together with the volume of the steam port that leads to it, is called

the **clearance.** This same term is also sometimes understood to mean the distance between the piston and the cylinder head at the end of a stroke, but this is not the usual or correct significance.

13. Piston. — The **piston** *P*, Fig. 3, is of cast iron or cast steel, and fits loosely in the cylinder. Its circumference is enlarged by split rings *k* and *k'*, called **piston-packing rings,** which press against the wall of the cylinder and thus prevent leakage of the steam between the cylinder wall and the piston. These rings *k* and *k'* may be held in place by the follower plate *h* that is bolted to the head end of the piston, or the piston may consist of a single casting provided with grooves into which the rings are sprung, doing away with the danger of accidents from a broken follower plate or follower bolts that hold the follower plate in place.

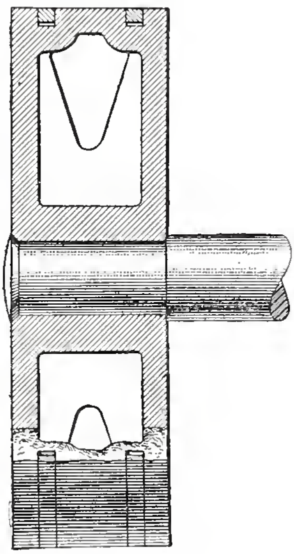


FIG. 5

Fig. 5 shows a simple piston much used for cylinders 20 inches or less in diameter. It is cast in one piece and is hollow to save weight. The end of the piston rod is riveted over after the piston is pressed on to it. The steam rings are sprung over the piston in grooves provided for them.

Fig. 6 illustrates a good piston for cylinders 22 inches and more in diameter. The main, or junk, ring on the outside of the piston is held in place by a follower ring *f* or a plate that is bolted to the piston. The steam rings *a* may be sprung into the junk ring. The piston rod is centered by two setscrews *s*.

The stuffingbox *f*, Fig. 3, in which packing is placed is fitted with a gland *J*, which when screwed up compresses the packing around the piston rod *l* and makes a steam-tight joint. When repacking, care should be taken not to cause unnecessary friction by too much pressure from the gland.

Drain (drip) cocks w and w' are fitted in each end of the cylinder, through which any water from the condensation of steam may be discharged.

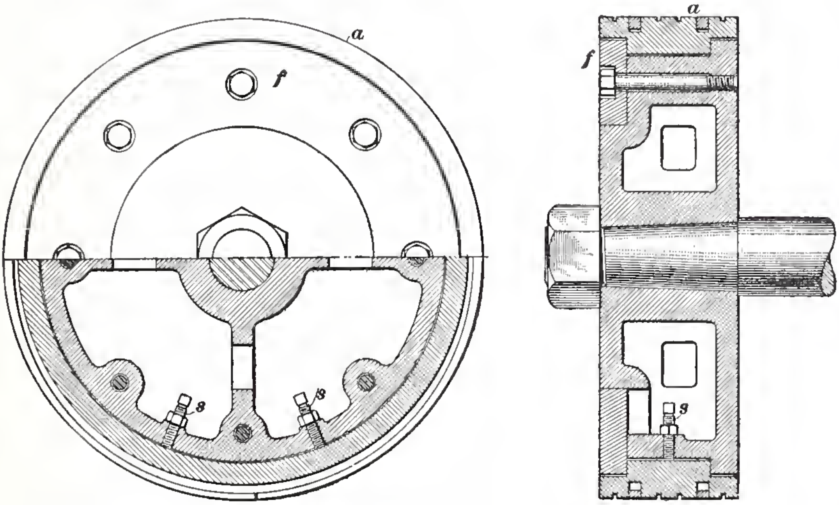


FIG. 6

14. The **crosshead** of an engine is usually made of cast iron or cast steel, and is fastened to the piston rod, which gives to it its to-and-fro motion. The shoes, or gibs, sliding in the guides compel the crosshead to move in a straight line; the crosshead pin furnishing a connection between the crosshead and the shoes, or gibs.

15. **Connecting-rods** are made of forged iron or steel, and are provided with adjustable boxes at each end working on the crosshead and crankpins. Their length is usually made from two and a half to three times the length of the stroke.

16. **Engine cranks** are made of cast iron, cast steel, and occasionally of forged iron or steel. They may be crank-shaped or of the disk pattern shown in Figs. 1 and 2; in the latter case there is usually a counterbalance weight on the side opposite the crankpin. Cranks should be pressed or shrunk on to the shaft and also keyed. Crankpins should

be pressed into the crank and riveted over on the end. Some makers key them as well.

17. The Eccentric.—Fig. 7 shows the eccentric that imparts motion to the slide valve *V*, Fig. 3. It consists of a circular disk of iron *a*, which, after being properly adjusted, is keyed or fastened by setscrews to the shaft that revolves with it. The center of this disk is at *O*. It is evident that, as the shaft revolves, the center *O* of the eccentric *a* will describe the dotted circle *b*, whose center is the center of the shaft. Consequently, the eccentric strap *c* and the

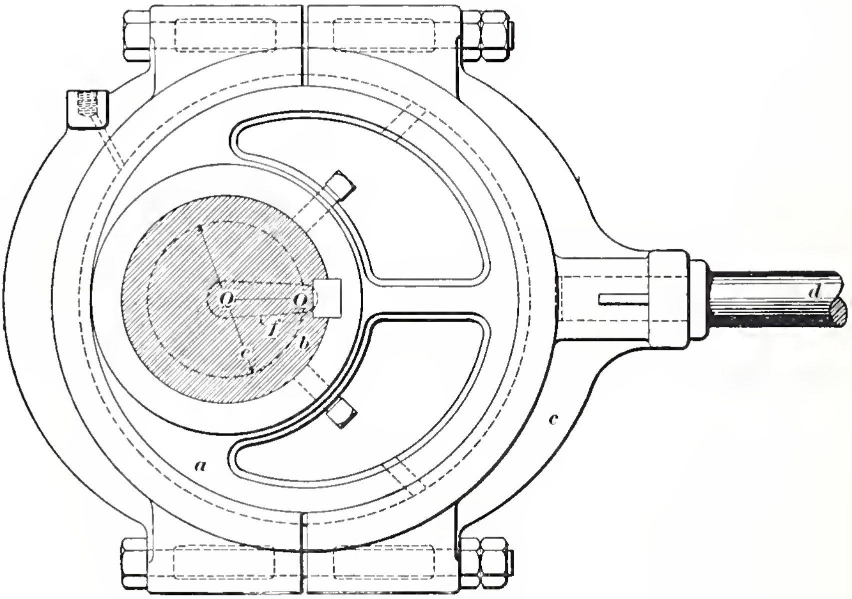


FIG. 7

eccentric rod *d*, to which it is fastened, will be moved, during a half revolution, a distance equal to the diameter *e* of the dotted circle. This distance *e* is the *throw* of the eccentric. The distance *O Q* between the center of the eccentric and the center of the shaft is the radius of the eccentric, or the *eccentricity*. It is plain that the throw is twice the radius. Practice varies somewhat in the definition of the term throw; some engineers call the radius the throw, but by far the greater number define throw as here given. The

eccentric is equivalent to a crank whose length is equal to the radius of the eccentric. Thus, if the end of the eccentric rod *d* were attached at *O* to the imaginary crank *f* (shown in dotted lines), the crank would give the same motion to the rod that the eccentric does.

THE SLIDE VALVE AND STEAM DISTRIBUTION

18. Fig. 8 shows a section of the slide valve, the section being taken in a plane passing through the center lines of

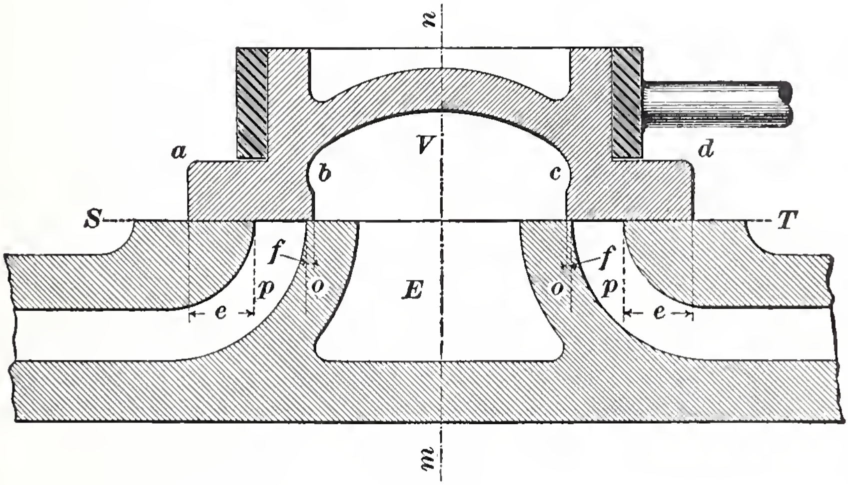


FIG. 8

the piston rod and valve rod. The valve is shown in mid-position, that is to say, the center line *n* of the valve coincides with the center line *m* of the exhaust port. Fig. 9 is a perspective view of a slide valve. In Fig. 8, *p*, *p* are the steam ports leading to the two ends of the cylinder; *o* and *o* are bridges; *E* is the exhaust port by which the steam leaves the cylinder. The opening *b c* is the throat of the valve; *S T* is the valve seat on which the valve moves

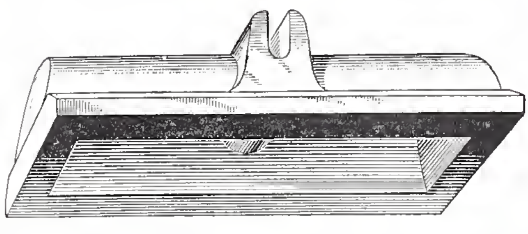


FIG. 9

to and fro, the motion being communicated to the valve by the valve stem or rod that receives its motion from the eccentric. As the valve moves to and fro, it alternately opens and closes the steam ports *p*, *p* and the exhaust port *E*, thus permitting steam to alternately enter and then exhaust

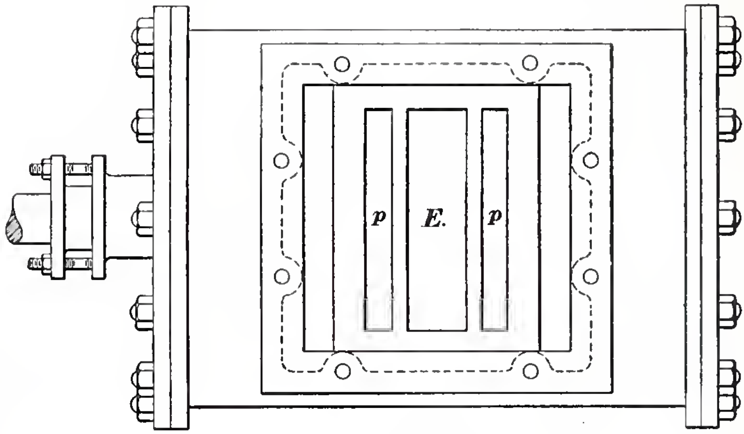


FIG. 10

from the ends of the cylinder. Fig. 10 is a plan of the valve seat showing the steam ports *p*, *p* and exhaust port *E*. The exhaust port opening is always made larger than the steam ports, so that the steam may escape freely.

19. Lap of Valve.—In Fig. 8, the flanges *a b* and *c d* of the valve are wider than the steam ports. The amount *e* that the valve overlaps the steam ports on the outside is the **outside lap**, and the amount *f* that the valve overlaps the ports on the inside is the **inside lap** of the valve. With an ordinary **D** slide valve, operated by one eccentric, there can be no cut-off, and therefore no expansion of steam, unless the valve has outside lap.

20. Effect of Lap.—The effect of lap on the movement of the valve relatively to the piston, and also on the movement of the eccentric and crank, is clearly shown in Figs. 11 to 18, in which the valve has both outside and inside lap. These diagrams have been distorted in order that the eccentric radius might be long enough to show up well.

In Fig. 11, the piston is represented as just beginning the forward stroke, and the valve as just commencing to open the left steam port, both moving in the same direction, as shown by the arrows. If the valve had no outside lap, the position of the eccentric center would be at *e*, but on

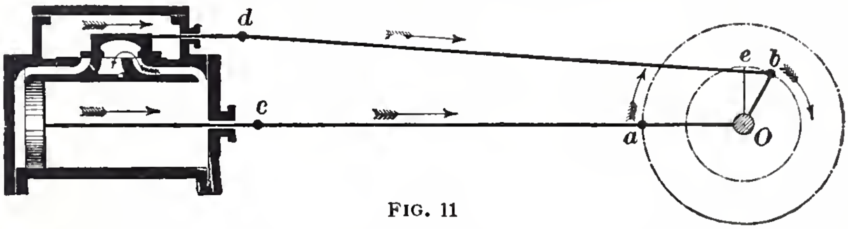


FIG. 11

account of the lap, the valve has moved ahead of its central position in order to bring its edge to the edge of the port. To accomplish this, the eccentric center has been moved from *e* to *b*, *O b* being the position of the eccentric radius. The angle *b O e*, which the eccentric radius makes with the position it would be in if there were no lap or lead, is called the **angle of advance.**

Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance space on the left of the piston, so that the steam acts on the piston at boiler pressure. The steam on the right side of the piston is flowing (exhausting) into the atmosphere through the exhaust port, as shown by the arrow. As the size of the exhaust port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston in addition to the atmospheric pressure. This is termed **back pressure.**

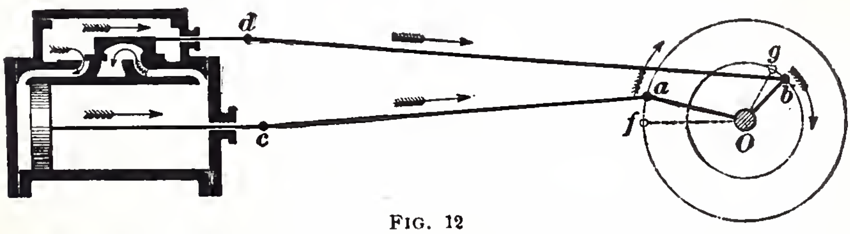


FIG. 12

21. Fig. 12 shows the position of the piston and valve when the exhaust port is fully opened. The crank has

moved from the position Of (shown by dotted line) to Oa, and the eccentric center from g to b. Steam is entering the head end of the cylinder and exhausting at the crank end.

22. In Fig. 13 the piston has advanced far enough to enable the valve to reach the end of its stroke and open the left-hand steam port its full width. The crank and eccentric have moved to the positions Oa and Ob, the dotted lines

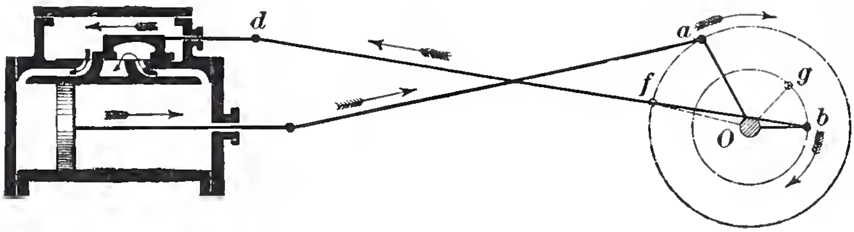


FIG. 13

showing their last position in Fig. 12. The eccentric radius is horizontal, and any farther movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back.

23. Fig. 14 shows the piston still farther advanced on its stroke, and the valve as having its inside edge in line with the outside edge of the exhaust port. The left end of

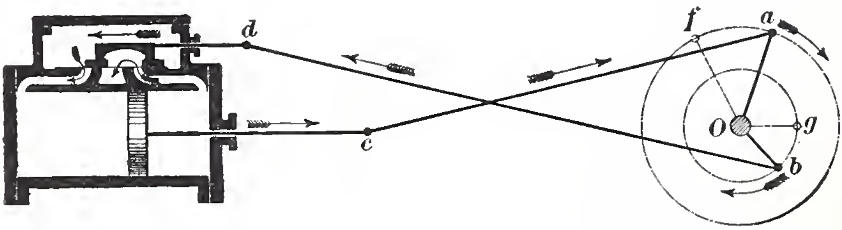


FIG. 14

the valve has partially closed the steam port. The amount of advancement of the crank and eccentric from their last positions is shown by their distances from the dotted lines.

24. Fig. 15 marks one of the most important points of the stroke. Here the valve has closed the steam port, i. e.,

cut off the steam. This point is called the **point of cut-off,** and from here to the end of the stroke the steam in the

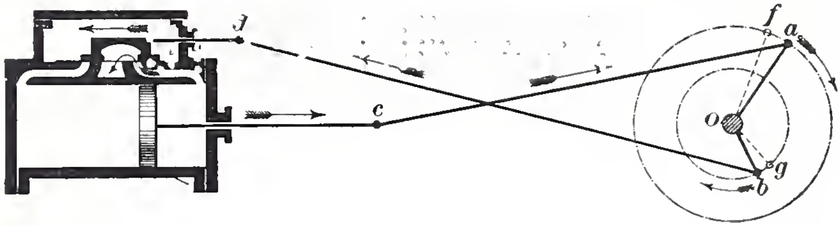


FIG. 15

cylinder expands. The exhaust port is now partly closed. The crank and eccentric have moved through the angles indicated.

25. When the inside edge of the valve closes the exhaust port, as shown in Fig. 16, the compression of the

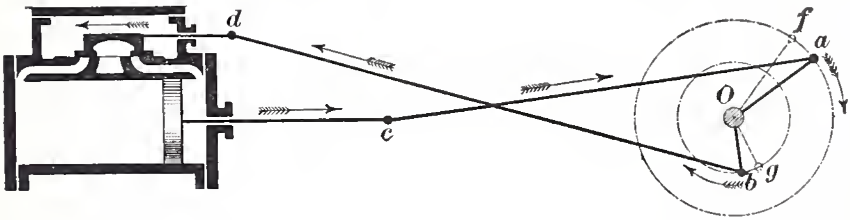


FIG. 16

steam in front of the piston begins. This point in the stroke is called the **point of compression,** and the time from then to the end of the stroke is the **period of compression.**

26. In Fig. 17 the piston has advanced far enough to

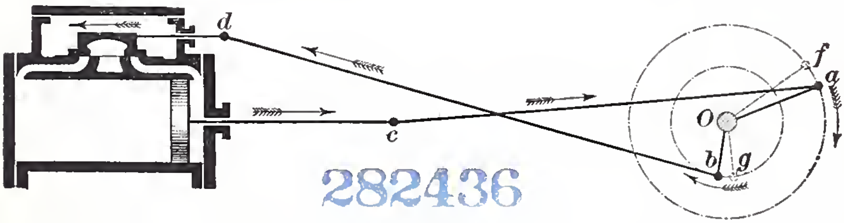


FIG. 17

cause the inside edge of the valve to be in line with the inside edge of the left port. The slightest movement of

the valve to the left will open the left port so that the steam in the left end of the cylinder will pass into the exhaust port. This point of the stroke is called the **point of release.** The work done by expansion theoretically ends here, although, on account of the limitation in the size of the ports, there will still be a slight further amount of work done by expansion, owing to the inability of the steam to escape instantly.

27. In Fig. 18 the piston has reached the end of its forward stroke and is about to begin the return stroke. The right outside edge of the valve is in line with the outside edge of the right port. The steam is exhausting from the head end of the cylinder, as shown by the arrows. The crank

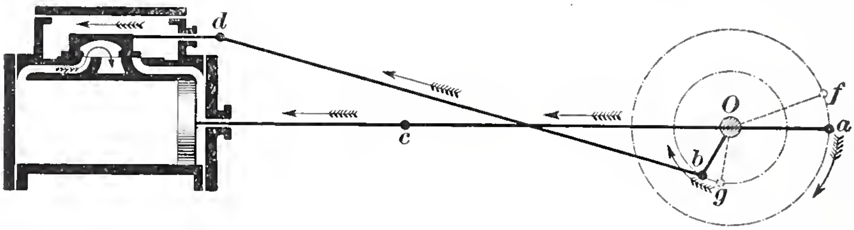


FIG. 18

and eccentric are both diametrically opposite their positions in Fig. 11, and the slightest movement of the piston to the left will admit steam to the crank end of the cylinder. During the return stroke, the actions of the steam described above will be repeated.

28. Relation of Lap to Expansion and Compression. In Fig. 15 it is evident that a greater outside lap would close the steam port earlier, and a less outside lap later in the stroke. In other words, increasing the outside lap means an earlier cut-off, and greater expansion of steam in the cylinder, together with economy in the use of steam. In Fig. 16 it is likewise evident that a greater inside lap would close the exhaust port earlier and open the same later in the stroke, which would cause compression of the steam remaining in the cylinder to begin earlier and release later. The effect of compression is to store up a portion of the energy of the moving parts of the engine at the end

of each stroke, by compressing the steam in front of the piston for a brief period. The energy so stored at the end of each stroke is at once given out at the beginning of the following stroke. Compression greatly assists in overcoming the inertia of the reciprocating parts of the engine, at the beginning and end of each stroke.

29. Lead.—A valve is said to have **lead** when it commences to open the steam port just before the piston reaches the end of the stroke. The width of the opening, say $\frac{1}{16}$ or $\frac{1}{8}$ inch when the piston has reached the end of its stroke, is the amount of lead. The amount of lead is measured by the distance between the edge of the valve and the edge of the

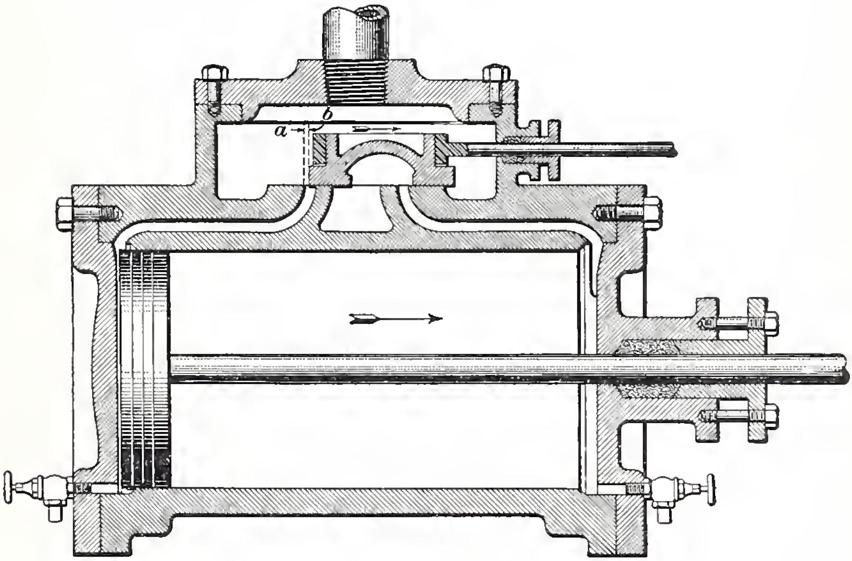


FIG. 19

port from which the valve is traveling. In Fig. 19, the lead is the distance *a b*. Most engineers give their valves lead in order to have the clearance space filled with live steam when the piston begins its stroke. This also adds to the quiet and smooth running of the engine, because the live steam aids the steam of compression in bringing the reciprocating parts to rest.

Since when a valve has lead, it is moved farther to the right than in the position shown in Fig. 11, it is evident that the center *b* of the eccentric must also be moved a little farther to the right; that is, to give a valve lead, the angle of advance must be increased. The effect of lead on the angle of advance of the eccentric is evidently the same as an increase of lap.

30. Position of the Eccentric.—Referring to Figs. 11 to 18, it is seen that when the valve has lap (or lap and lead), the angle *a O b* between crank and eccentric is greater than 90°. Following the direction of the arrows, it is seen, however, that the eccentric *b* reaches the lowest point on the circle earlier than the crank *a* reaches the lowest point on its circle; that is, the eccentric is *ahead* of the crank. Take now the case of an engine that runs under, as shown in Fig. 20. The crank is in position *a* and is about to move

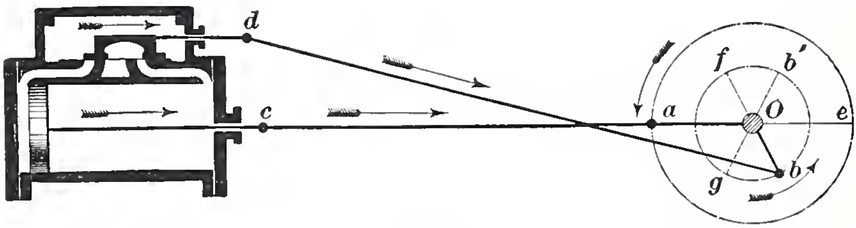


FIG. 20

downwards. The eccentric must be in the position *O b*. An inspection of the diagram shows that, following the direction of the arrows, the eccentric is set *ahead* of the crank, and the angle between the crank and eccentric is $a\,O\,b = 90° +$ angle of advance.

Hence, for the ordinary slide valve, the following general rule applies:

Rule.—*When the valve rod and eccentric rod move in the same direction, the eccentric is set* **ahead** *of the crank, and the angle between the crank and eccentric is 90° + the angle of advance.*

This is true whether the engine runs over or under.

31. Rocker-Arms.—It frequently happens that the eccentric cannot be located on the shaft so that the eccentric rod and valve stem shall be in the same straight line. Again, it is sometimes desirable to make the throw of the eccentric less than the valve travel. In such cases a lever, or **rocker-arm,** Figs. 21 and 22, may be pivoted to a fixed

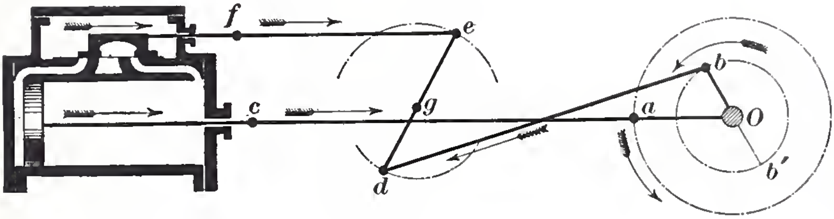


FIG. 21

point of the engine bed, and the valve and eccentric rods connected to it. In Figs. 21 and 22, *g* is the fulcrum, or fixed point; *d* is the point at which the eccentric rod is attached, and *e* the point at which the valve rod is attached. By varying the lengths of the lever arms *gd* and *ge*, the valve stem may be made to travel a greater or less distance

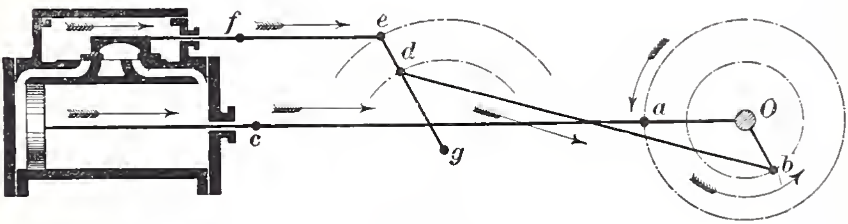


FIG. 22

than the throw of the eccentric. If the rocker-shaft is constructed so as to have an arm each side of the pivot, Fig. 21, the valve will move in a contrary direction to that of the eccentric, but if the valve stem and eccentric rod are fastened to the shaft arm on the same side of the pivot, Fig. 22, both the valve and eccentric rod will move in the same direction.

32. Direct and Indirect Valves.—A slide valve is said to be **direct** when it opens the left port by moving to the right and closes it by moving to the left. A valve is said to be **indirect** when it opens the left steam port by moving to the left and closes it by moving to the right. The plain slide valve is a direct valve. It opens the left port by moving to the right, admits steam past the outside edge, and exhausts it past the inside edge.

The **piston valve**, shown in Fig. 23, is the most familiar form of indirect valve. It consists of a hollow cylinder sliding on a cylindrical valve seat. The ports *P* extend clear around the valve. The steam is admitted into the central chamber *A*, and the exhaust steam escapes out at the two ends *B*. As shown in the figure, the piston is just about to start to the right, and the valve is moving to the

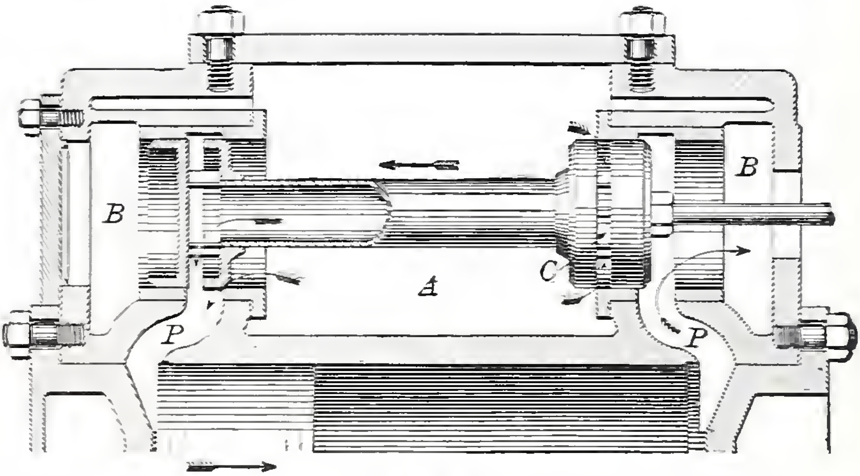


Fig. 23

left, thereby uncovering the left steam port and allowing the steam to enter past its inside edge. The valve is, therefore, indirect. To give a larger admission, steam also passes into the center of the valve through the channel *C*, and thence into the left port. The exhaust steam meanwhile escapes directly through the right steam port into the chamber *B*.

33. Position of Eccentric for Indirect Valve.—Since the direction of motion of an indirect valve is precisely opposite that of a direct valve, the eccentric must be set exactly opposite the position it would have were a direct valve used. The following rule will give the position of the eccentric:

Rule.—*When an indirect valve is used, set the eccentric* **behind** *the crank and make the angle between them equal to 90° minus the angle of advance. If a rocker is used, that makes the valve rod and eccentric rod move in opposite directions, then set the eccentric* **ahead** *of the crank and make the angle between them equal to 90° plus the angle of advance.*

This rule applies whether the engine runs under or over.

34. The **Meyer cut-off valve**, Fig. 24, is used extensively in air compressors. It cuts off the steam early in the stroke. With the plain slide valve, the cut-off cannot take place much, if any, before the engine has made half a stroke.

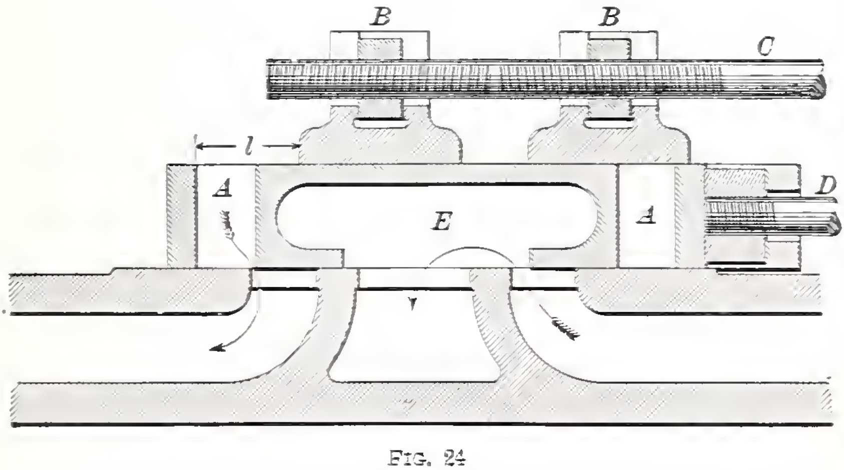


FIG. 24

If the valve is given enough lap to cause it to cut off early, compression must necessarily be early also. This is readily seen by examining Figs. 11 to 18. The Meyer valve consists simply of a flat plate. The steam ports A pass through

the valve. The lower side of the valve contains the hollow, or throat, E to receive the exhaust. On top of this main valve slides the cut-off valve, which consists of two plates B, B moved by the valve rod C. The main valve is moved by the valve rod D. The action of the valve may be seen from Fig. 25. Here $O\,a$ represents the crank position as the piston is about to make the forward stroke. The eccentric $O\,b$ of the main valve is set in its usual position $90°$ + angle of advance ahead of the crank. The eccentric $O\,c$ is set $180°$, or a little more, ahead of the crank. Now, when the crank moves in the direction shown by the arrows, the eccentric b moves to the right and eccentric c moves to the left. Hence the main valve moves to the right and the plates B to the left, and the distance l, Fig. 24, between the left edge of the passage A and the left edge of the plate B decreases. When l becomes zero, that is, when plate B completely covers passage A, the steam is cut off, no matter what the position of the main valve may be. The cut-off valve only affects the point of cut-off, the lead, compression, and release being regulated by the main valve the same as with the simple **D** valve.

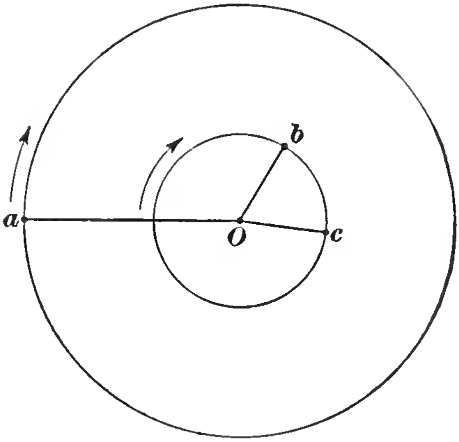


Fig. 25

The rod C and the plates B, B are provided with right-handed and left-handed screw threads, so that by rotating the rod the plates may be spaced closer together or farther apart. In the former case, the cut-off is made to take place later; in the latter case earlier. As usually arranged, the rod C may be turned by a hand wheel placed outside the steam chest, so that the cut-off may be changed while the engine is running, the mechanism being graduated to show the point of cut-off.

SETTING THE SLIDE VALVE

35. Dead Centers.—Fig. 26 shows that when the piston is at the end of its stroke at the end *h* of the cylinder, the crankpin must be at the point *m* in the crankpin circle. In

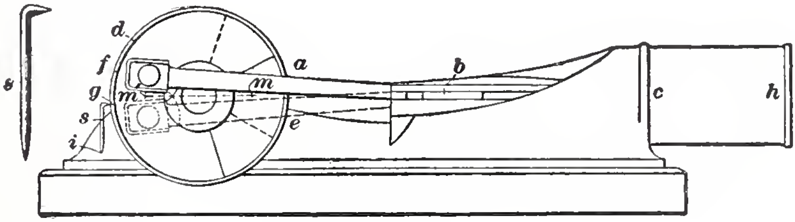


FIG. 26

this position the crank and connecting-rod lie in the same straight line. Likewise, when the piston is at the other end of the stroke, the crankpin lies at the point *m'*, and again the crank and connecting rods are in the same straight line.

When the crank occupies either of these positions, the engine is said to be on its **dead center.** All the pressure of the steam on the piston is transmitted directly to the shafts, because the reciprocating parts are in a straight line. Consequently, there is no tendency to turn the crank, and the engine cannot be started until turned into a different position. When the crank occupies the position *m*, it is said to be on its *interior* dead center, and when it occupies the position *m'* on its *exterior* dead center.

36. Placing the Engine on Dead Center.—It is often necessary to place the crank on the dead center when setting the valve, and this is done in the following manner: The crank is turned so that the connecting-rod will stand in the position shown by the full lines *a*, Fig. 26, and a line *b* is drawn across the crosshead and guide. A scriber, or tram, *s* should be placed in a prick mark *i* on the engine bed and a line *g* drawn on the crank disk. The crank should now be rotated so as to bring the connecting-rod into the

position *e*, shown by the dotted lines, when the lines *b* on the crosshead and guide will again coincide; another line *d* is then drawn on the disk crank. The distance from *g* to *d* is bisected with a pair of dividers, giving the line *f* on the disk crank; now move the crank until this middle point *f* coincides with the end of the tram, when the engine will be on the dead center. This operation may be repeated when it is desirable to get the dead center on the other end of the stroke. If the crank is of such form that it is not convenient to use the tram on it in this manner, the marks may be made on the flywheel. To avoid error that may come from lost motion, turn the engine toward the center when the mark is made and when setting to the mark *f* by the tram, so that the connecting-rod brasses will press on same side of crank and crosshead pins.

37. General Consideration of Valve Setting.—Only a general rule can be given for setting the valves of steam engines, as the work is largely a matter of judgment. The plain slide valve is the one most commonly met, and a description of the manner of setting this will be given. As the valve gear is generally constructed, there are two adjustments provided. The first consists of a change in the length of the valve stem or of the eccentric rod, and the second of rotating the eccentric on the shaft.

38. To Center the Valve.—The valve is made to travel equally each way from mid-position by altering the length of the valve stem. For example, if the valve travels $\frac{1}{2}$ inch too far toward the head end, shortening the stem one-half that distance pulls the valve $\frac{1}{4}$ inch nearer the crank and makes it travel equally each way. Any rotation of the eccentric on the shaft hastens or retards the valve action as it may be moved ahead or back.

39. To Set a Slide Valve.—Place the engine on either dead center by rotating the crank-shaft in the direction in which it is to run. Then, if the eccentric rod is directly connected to the valve stem, that is, if the direction of

motion of both the eccentric rod and valve stem is the same, move the eccentric on the shaft a little more than a right angle ahead or in advance of the crank. But if the eccentric rod is cross-connected to the valve stem, that is, if its direction of motion is directly opposite to the motion of the valve stem, place the eccentric on the shaft a little less than a right angle behind or following the crank. Continue to increase or decrease the angle between the eccentric and the crank by turning the eccentric on the shaft until the valve has the desired lead; that is, until it is open, say from $\frac{1}{16}$ inch to $\frac{1}{8}$ inch, then temporarily fasten the eccentric to the shaft.

Again turn the engine in the direction it is to run until it is on the other dead center. If the lead is the same as at the other end, the valve is correctly set; if it is not the same, the valve rod must be lengthened or shortened until the lead is the same at both centers. If the lead is less than desired. turn the eccentric forwards on the shaft for a direct-connected valve and backwards for a cross-connected valve; if the lead is too great, turn the eccentric in the opposite direction. When the lead is right, fasten the eccentric securely to the shaft.

After the valves are set and the engine is started, a pair of indicator diagrams (to be described later) should be taken. The diagrams will show any slight errors in the setting.

40. To Find Point of Cut-Off.—Slide-valve engines are made to cut off at different points in the stroke, according to the conditions under which they are to be operated. The point of cut-off is expressed in the form of a ratio. Thus, if the length of stroke is 36 inches, and the steam is cut off when the piston has travelled 24 inches, the valve is said to cut off at $\frac{2}{3}$ ($\frac{24}{36}$) stroke. The ratio of expansion is the ratio between the length of the stroke and that part of the stroke before the cut-off takes place.

Take the case of a 48-inch-stroke Corliss engine, with a given load, cuts-off of 12 inches, then the valve has $\frac{1}{4}$ cut-off and the ratio of expansion is $48 \div 12 = 4$. In practice, the point of cut-off can be obtained directly from the engine. Mark the dead-center points on the guide bars and measure

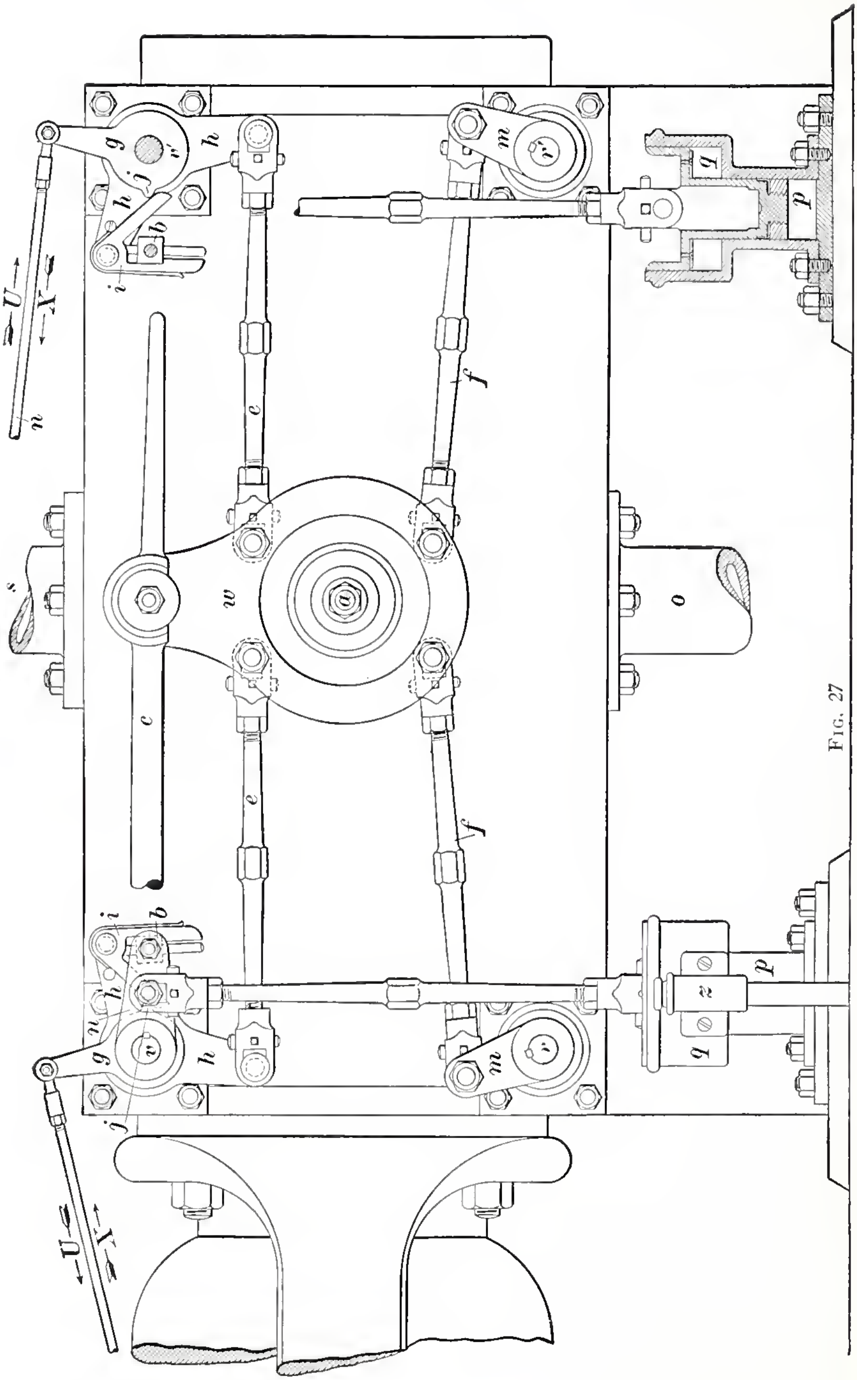


FIG. 27

the distance between them; this will be the length of the stroke. Take off the steam-chest cover, and with the piston at the head end of the cylinder, slowly rotate the engine forwards until the head edge of the slide valve in closing the steam port just reaches the head edge of the steam port. Then measure the distance between the head dead-center guide-bar mark and the mark on the crosshead; divide this latter quantity by the one first taken, and the result will be the cut-off required, in a fraction of the stroke. Plain slide valves usually cut off between $\frac{1}{2}$ and $\frac{3}{4}$ stroke; Corliss valves usually cut off between $\frac{1}{4}$ and $\frac{1}{2}$ stroke.

CORLISS VALVE GEAR

41. General Description.—The plain slide valve involves most of the principles in the more complicated forms of valve gear at present in use. There are several

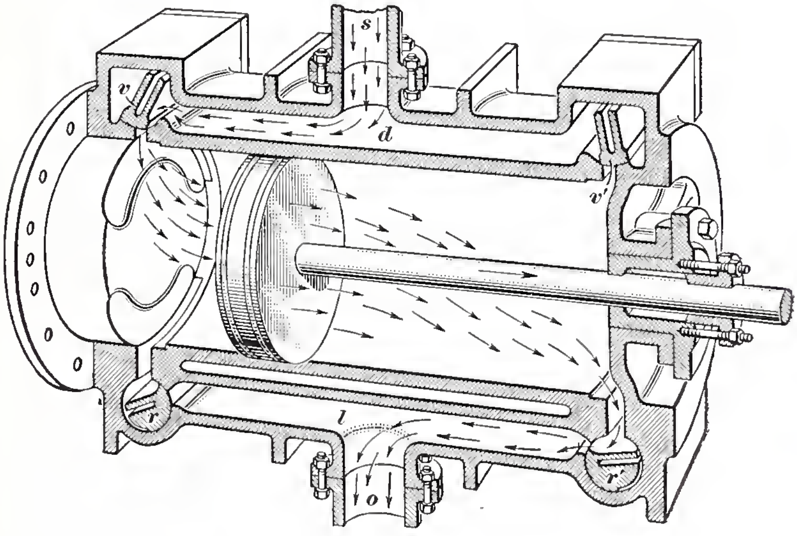


FIG. 28

forms of automatic valve gears in use. The **Corliss valve gear** is, however, being so extensively employed that a short description of its working parts and principles is here given.

Fig. 27 shows a side elevation of this valve gear, and Fig. 28 a section through the cylinder and valves. It has four separate and distinct valves. Two of these *v* and *v'*, Fig. 28, called steam valves, connect directly with the steam chest *d* and steam pipe, or throttle, *s*. They are rigidly connected with the cranks *n*, Fig. 27. The right-hand crank is removed in order to show more clearly the disengaging hook *i*. The other two valves *r* and *r'*, Fig. 28, called exhaust valves, connect directly with the exhaust chest *l* and the exhaust pipe *o*; they are rigidly connected with the cranks *m*, Fig. 27. All the valves are cylindrical in form, and extend across the cylinder above and below, respectively.

The wristplate *w* is made to rock on a stud *a*, by the hook rod *c*, connecting it with an eccentric on the crank-shaft. Two motion rods *e* connect the wristplate *w* with the bell cranks *h* of the steam valves, and two motion rods *f* connect the wristplate with the cranks *m* of the exhaust valves. The motion rods can be lengthened or shortened as the case may require, and the action of any one valve may be regulated independently of the other three. As the wristplate *w* rocks backwards and forwards, the exhaust valves *r* and *r'*, which are rigidly connected to their cranks *m*, rock with it. The bell cranks *h*, which are provided with the disengaging hooks *i*, are also given this rocking motion, and by hooking on to the blocks *b*, which are rigidly connected to the cranks *n*, open the steam valves *v* and *v'*.

42. The projections *j* on the two trip collars *g* unhook the disengaging hooks *i* after they have rotated the valves *v* and *v'* through a certain angle, regulated or determined by the governor through the rods *UX*. The cranks *n* are pulled back to their first positions by the vacuum dashpots *p* against the resistance of which the valve cranks *n* were raised. The drop is cushioned by the air in *q*, the escape of the air being regulated by the valve *z*. The movements of the valves open and close the steam and exhaust ports of the cylinder at the proper intervals. The pins of the motion rods are so

located on the wristplate that the steam valves v and v' have their quickest movement while the exhaust valves r and r' have their slowest movement, and the exhaust valves have their quickest movement while the steam valves have their slowest movement. As a consequence of this arrangement, the steam and exhaust valves have entirely independent movements, and the inlet ports may be suddenly opened full width by the quick movement of the steam valves, while the exhaust valves are practically motionless. The advantage of this valve gear is that it permits an earlier cut-off with a greater range and a more perfect steam distribution than is attained with the plain slide valve. Engines fitted with the Corliss valve gear do not usually run at much more than 100 revolutions per minute, as they will not hook and unhook to the steam valves and dashpots with certainty beyond that speed, although Corliss engines of special design have been run 160 revolutions per minute.

SETTING CORLISS VALVES

43. Centering Rocker-Arm and Wristplate.—In most cases three marks c, b, d, Fig. 29, are on the wristplate bracket and another mark a on the hub of the wristplate, by means of which the wristplate may be centered. The marks are so located that a is opposite b when the wristplate is at its center of motion. At the two extremes of motion a is opposite either c or d. It may be well, however, to test these marks, or rather to see that the eccentric and rods have proper adjustment relative to the motion of the wristplate. To do this, rotate the eccentric p, Fig. 30, on its shaft; then notice whether or not the rocker-arm r is equidistant in its extreme travel each way from a plumb-line let fall through the center of its pin. If it is not, make it so by adjusting the length of the eccentric rod q. Next, see if the mark on the wristplate hub agrees

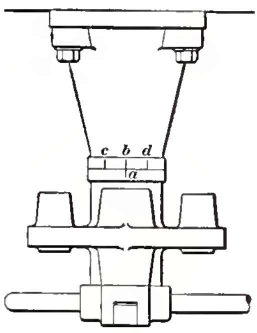


FIG. 29

with those on the bracket at full throw each way; if not, the remedy is to change the length of the hook rod *c* until

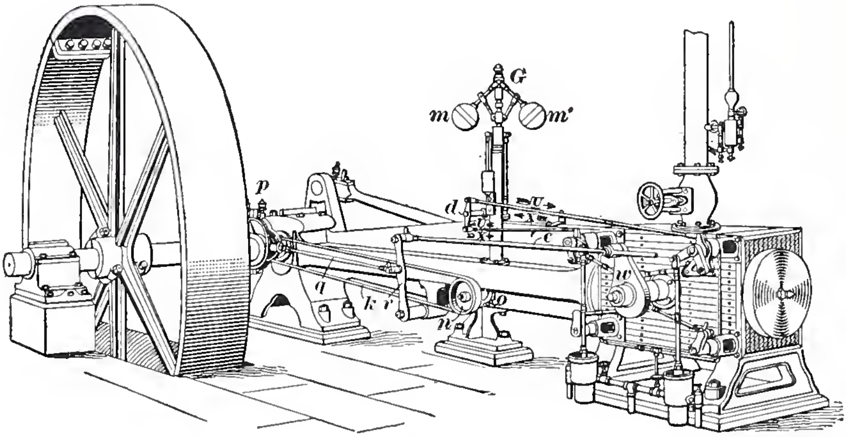


FIG. 30

there is perfect agreement. Having tested the marks temporarily, secure the wristplate *w* at its center of motion.

TABLE I

STEAM LAP AND EXHAUST OPENING FOR CORLISS ENGINES

Diameter of Cylinder Inches	Lap for Steam Valve Inches	Exhaust-Valve Opening Inches	Diameter of Cylinder Inches	Lap for Steam Valve Inches	Exhaust-Valve Opening Inches
12	$\frac{1}{4}$	$\frac{1}{32}$	30	$\frac{1}{2}$	$\frac{1}{8}$
14	$\frac{5}{16}$	$\frac{1}{32}$	32	$\frac{1}{2}$	$\frac{1}{8}$
16	$\frac{5}{16}$	$\frac{1}{32}$	34	$\frac{1}{2}$	$\frac{1}{8}$
18	$\frac{3}{8}$	$\frac{1}{16}$	36	$\frac{1}{2}$	$\frac{1}{8}$
20	$\frac{3}{8}$	$\frac{1}{16}$	38	$\frac{9}{16}$	$\frac{3}{16}$
22	$\frac{3}{8}$	$\frac{1}{16}$	40	$\frac{9}{16}$	$\frac{3}{16}$
24	$\frac{7}{16}$	$\frac{3}{32}$	42	$\frac{9}{16}$	$\frac{3}{16}$
26	$\frac{7}{16}$	$\frac{3}{32}$	44	$\frac{5}{8}$	$\frac{1}{4}$
28	$\frac{7}{16}$	$\frac{3}{32}$	46	$\frac{5}{8}$	$\frac{1}{4}$

44. Amount of Lap.—No definite rule can be given by which the amount of lap for a valve can be determined. It depends somewhat on the design of the valve and its relative proportions; also on the conditions under which the engine is to work. In all cases the lap increases with the size of the cylinder. Table I furnishes a fairly reliable guide as to the amount of lap to be given to valves on different sizes of engines.

45. Adjusting the Lap.—On removing the back bonnets, or caps, from the ends of the valve chambers, so that the rear ends of the valves are exposed, a mark will usually be found on each face of the valve ports, showing the location and width of the port openings. On the ends of the valves, Fig. 31, marks in line with the opening edges of the valves will also be found. Possibly in some of the older types of engines these may be missing; in such a case the valves will have to be removed to locate the port openings and the opening edges of the valves. Consulting Table I, find for the engine being adjusted the lap for the steam valves and the opening to be given the exhaust valves. By lengthening or shortening the motion rods leading from the wristplate to the valve arms, the opening edges of the valves may be adjusted to correspond with the desired lap.

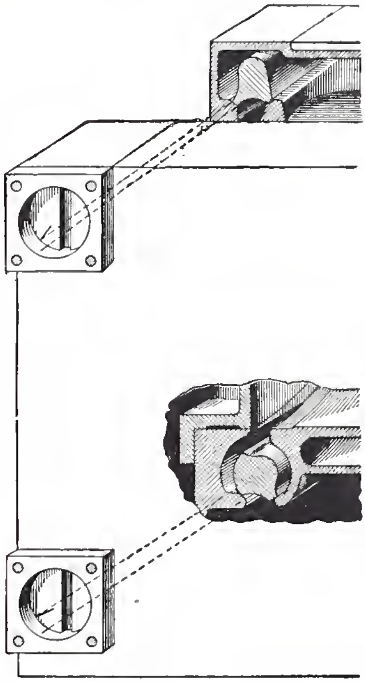


FIG. 31

If a record is kept of the distance the valve moves at one turn of the adjusting nut on the rod, future adjustments may be made without the removal of the bonnets.

46. Adjusting the Lead.—All the valves are now supposed to be in their proper positions when the wristplate is at its center of movement. The next step is to locate the eccentric at the proper angle ahead of the crank to give sufficient lead. First, set the engine exactly on its center, and revolve the eccentric on the shaft in the direction in which the engine is to run, until it is at an angle greater than 90° ahead of the crank, or until the steam valve on the end at which the piston stands is just beginning to open. Some give from $\frac{1}{32}$ inch to $\frac{1}{16}$ inch opening as lead. In this position the eccentric must be secured to the shaft. Then turn the engine to the other center and see if the steam valve on that end has the same amount of opening as the other had. It should and will have the same amount, if all adjustments have been carefully made.

The valves, Fig. 28, are the common Corliss valves opening on one edge, thus allowing steam to pass to one side of the valve only, as shown. Fig. 31 shows Corliss valves opening on two edges, their shape being such that when open, steam passes into the ports on each side of the valves.

47. Adjusting the Knock-Off Rods.—For the purpose of adjustment, block the governor *G*, Fig. 30, so that the balls *m*, *m'* stand in the position they would assume at normal speed (about mid-position) and fasten the knock-off rod lever *d* so that it will stand perpendicular. Now turn the engine to the point at which cut-off should occur (usually about $\frac{1}{5}$ stroke) and adjust the knock-off rod for that end, so the valve will trip at that point. The valve and the knock-off rod for the other end of the cylinder must be adjusted in a like manner. To determine the point of $\frac{1}{5}$ stroke, mark the length of stroke on the crosshead guides and measure off $\frac{1}{5}$ of this from each end. After a few trials, partially rotating the engine back and forth, at the same time making careful adjustments of the knock-off rods, cut-off can be made to take place at exactly similar points for each end. It is well, now, to drop the governor to its lowest position and observe that the cut-off mechanism does not

work, but allows steam to enter the cylinder during nearly the full stroke of the piston. To regulate the sensitiveness of the governor, vary the amount of opening between the two ends of the dashpot cylinder on the governor.

48. Dashpot Adjustments.—Care must be taken, in making adjustments of a valve gear, not to overlook even the smallest detail. For instance, the dashpot rods must have such a length that the steam arm will be in a position where the hook will surely catch on when the dashpot plunger is down.

49. General Summary.—To regulate the point of cut-off so that the same amount of steam is admitted at both ends, adjust the lengths of the knock-off rods *U*, *X*, Fig. 27. To give more or less steam lap and lead, lengthen or shorten the steam rods *e*, *e*. A change in the exhaust rods *f*, *f* likewise affects the cushion and release. After the eccentric has once been properly set, it is not necessary to disturb it in ordinary cases. If the dashpot rod is too short, the hook will not take hold. Look out for this. It is an excellent plan to mark the positions of the several parts where they are in adjustment so that one can tell at a glance if any adjustments have been disturbed. This marking may be done by making a fine punch mark on each rod 2 inches from each locknut on all the adjustable rods, such as the eccentric rod, hook rod, motion rod, dashpot rods, and knock-off rods. Then, if a nut works loose and has to be tightened up, the adjustment that would be affected by the loosening of the nut can be tested by simply measuring the distance from the punch mark to the nut. As a final test of the valve setting, indicator diagrams are to be recommended.

CORLISS GEARS WITH TWO ECCENTRICS

50. In order to secure a satisfactory steam distribution when the steam and exhaust valves of a Corliss engine are both driven by the same eccentric, it is necessary to give the eccentric a certain angle of advance. The releasing

gear cannot act on the valves after the eccentric has passed its extreme throw position. With a single eccentric, the releasing gear cannot be made to act after the piston has passed through a fraction of its stroke always less than $\frac{1}{2}$ and often little more than $\frac{3}{8}$. If the load on the engine is so heavy that cut-off does not take place before this fraction of the stroke has been completed, the steam valves will not be under the control of the governor, but will be closed only by the action of the eccentric.

The range at which cut-off can be controlled by the governor can be considerably extended by the use of two eccentrics and wristplates—one for the steam valves and one for the exhaust valves.

51. Amount to Which the Range of Cut-Off Can Be Economically Extended.— By setting the steam eccentric at a small enough angle with the crank, the range of cut-off with a double eccentric might be extended to nearly or quite full stroke. Such a range, however, is not desirable; it is not economical to use an engine under a load that makes it necessary for steam to follow the piston at boiler pressure for such a large part of the stroke, and, in addition, an attempt to secure a long range of cut-off by setting the eccentric back near to the crank position will cause the steam valves to open very slowly at the beginning of the stroke, thus producing wiredrawing of the steam and reducing the economy when working under normal loads. Satisfactory results can be obtained with the eccentric set at an angle of as little as 81° with the crank, 9° back of the 90° position; this provides for an extreme range of cut-off of about $\frac{7}{10}$ stroke.

STEAM-ENGINE GOVERNORS

52. Steam-engine governors are mechanical devices that regulate the steam supply of an engine, so that when the load on the engine is increased or decreased, or when the steam pressure under which it operates changes, the speed of the engine will remain practically constant. The

duty, however, of adjusting the working conditions of an engine to any sudden variation of steam pressure or load does not rest on the governor alone. It is the office of the flywheel alike to respond to these rapidly changing conditions, and by its inertia prevent any rapid change in the velocity or speed of the engine. When the engine is not supplied with a flywheel, there are other rotating parts, such as the drum of a hoisting engine, that serve the same purpose. Any variation of the speed of the flywheel is at once met by the action of the governor, which so alters the steam supply as to maintain a uniform velocity of the flywheel.

53. Steam-engine governors may be divided into two classes: (1) *throttling governors*, which throttle the steam in the supply pipe, and (2) *automatic cut-off governors*, which regulate the steam supply by changing the point of cut-off of the valve.

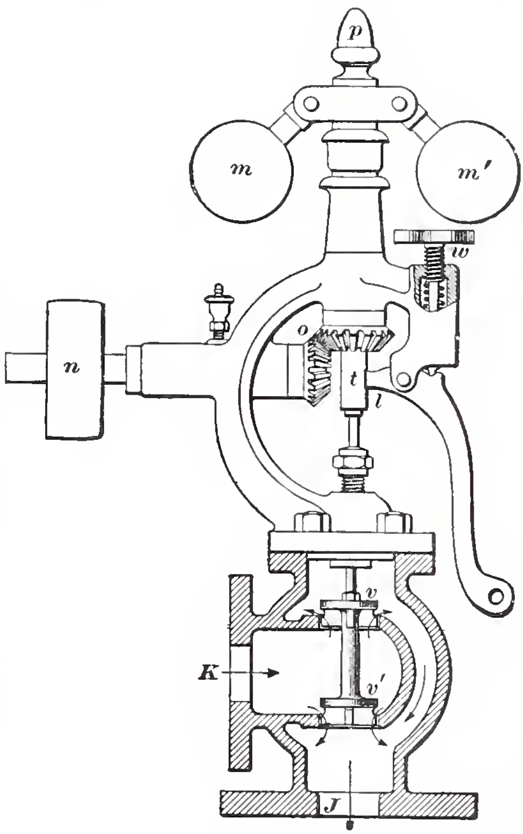


FIG. 32

54. Throttling governors are usually of the pendulum or fly-ball type *m*, Fig. 32. When the engine is running, a rotary motion is given to the pulley *n* by means of a belt that runs over a pulley rigidly fastened to the crank-shaft. This motion is transmitted through the bevel gears *o* to the spindle *p*, to which are secured the balls *m* and *m'*.

Suppose that the engine is running at its proper speed; a balance will then exist between the gravity force and the centrifugal force due to the rotary motion, both of which are acting on the balls. If, now, from any cause, the speed lessens, the centrifugal force will diminish, and gravity, acting on the balls, will pull them down. This movement on the part of the balls will, in turn, be imparted to a balanced throttle valve *v*, which will be opened wider, causing an increase in the initial pressure of the live steam passing into the cylinder. The live steam will now, in consequence of the additional amount of work it is capable of doing, exert more energy on the piston and bring the speed back to its proper point. If, on the other hand, the speed be increased, the balls will rise in consequence of the centrifugal force becoming greater than the attraction of gravity, and the steam orifice of the throttle valve will be diminished in area. This will lower the initial pressure; the steam will consequently exert less energy, and the speed will drop to its proper point.

55. Automatic Cut-off Governors. — Referring to Fig. 30, it may be understood that a rotary motion is imparted to the balls *m* and *m'* by means of a belt *k*, pulley *n*, and bevel-gear connection *o*, similar to that already described when stating the principle of the throttling governor. Suppose that the engine is running at its proper speed. The balls *m*, *m'* will then be held in their normal position by the balance existing between the centrifugal and gravity forces. Suppose, now, the speed of the engine increases from any cause whatever; the centrifugal force will increase and will continue to force the balls out, that is, to increase the diameter of the circle in which they rotate. This movement of the balls will be transmitted to the lever *d*, causing it to turn slightly about its center in the direction of the arrow *X*, Figs. 27 and 30. The movement of *d* will cause the trip collars *g*, Fig. 27, to turn through a small angle in such a direction that their projections *j* will disengage the hooks *i* and cause the point of cut-off to occur earlier in the

stroke, thus decreasing the speed of the engine on account of the reduction in the amount of steam admitted to the cylinder. Should the speed diminish from any cause, a reverse operation would be the result of the action of the governor, and the balls would drop slightly; *d*, Fig. 30, would turn as indicated by the arrows *U*, the trip collars *g*, Fig. 27, would be rotated in such a manner as to cause their projections *j* to unhook the disengaging hooks *i*, so that the cut-off would occur later in the stroke and the steam following the piston longer would bring the speed up to the proper point again. By tightening the speeder screw *w*, Fig. 32, the lever end *l*, the spindle *t*, and the balls *m* are raised and the speed of the engine increased.

INDICATOR AND INDICATOR CARDS

56. The **indicator** is an instrument that may be attached to the steam cylinder of an engine to determine the actual pressure on the piston at all points of its stroke. It also gives information as to working conditions, such as the operation of valves, piston, and other parts of the engine.

By means of the indicator, an **indicator diagram,** or **card,** may be drawn on which vertical distances represent pounds pressure per square inch and the horizontal distances represent the position of the piston in its stroke.

The indicator, as shown in cross-section, Fig. 33, consists of a cylinder *a* containing a piston *g* and a spiral spring *d*. It is attached to the engine cylinder by suitable pipes supplied with a cock, by means of which steam may be admitted to or shut off from the cylinder *a*. When steam is admitted through *s*, its pressure causes the piston *g* to rise. The spiral spring *d* is compressed, and resists the upward movement of the piston. The height to which the piston rises is in proportion to the pressure of the steam, and, as the steam pressure rises and falls, the piston rises and falls accordingly. To register this pressure, a pencil *p*

is attached to the end of the rod *n p*, the point of the pencil being made to press against a piece of paper, placed about

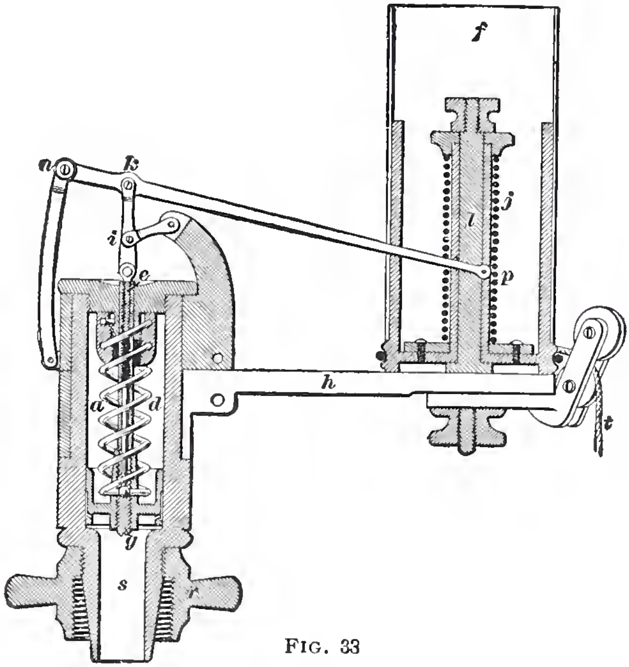


FIG. 33

the drum *f*, Fig. 34. As the piston *g* is moved up and down by the variable steam pressures in the engine cylinder, the pencil likewise moves up and down on the indicator drum *f*, making marks on the paper, which vary in length as the pressure varies. The drum *f* is rotated at the same time an equal amount for each stroke of the engine piston. It is desirable to restrict the maximum travel of the indicator piston to about $\frac{1}{2}$ inch; while the height of the diagram traced by the pencil may advantageously be 2 inches or more. To obtain a long pencil movement combined with a short travel of the indicator piston, the pencil is attached at *p* to the long end of the lever *n k p*.

57. Indicator Springs.—The height to which the piston will rise under a given steam pressure depends on the stiffness of the spring *d*, Fig. 33. Indicators are usually furnished with a number of springs of varying degrees of

stiffness, which are distinguished by the numbers 20, 30, 40, etc. These numbers indicate the pressure per square inch required to raise the pencil 1 inch. Thus, if a 40 spring is used, a pressure of 40 pounds per square inch raises the pencil 1 inch, and the vertical scale of the diagram is, therefore, 40 pounds per inch; that is, the vertical distance in inches of any point on the diagram, from the atmospheric line, multiplied by 40 gives the gauge pressure per square inch at that point. The scale of the spring chosen should not be less than half the boiler pressure, or the spring may be overloaded. For example, a 40 spring should be taken for a steam pressure of 75 pounds per square inch. If, on trial, the spring chosen makes a wavy line, choose a stiffer one. A stiffer spring is required on a fast-running engine than on a slow engine when the steam pressure is the same.

The indicator must not merely register pressures, but it must register them in relation to the position of the engine piston. To accomplish this, the drum *f* is revolved on its axis *l*, Fig. 33, by pulling the cord *t* that is coiled around it. When the pull is released, the spring *j*, Fig. 33, rotates the drum back to its original position. If the cord *t* be attached to some part of the engine that has a motion proportional to the motion of the piston, the motion of the drum will also be proportional to the motion of the piston.

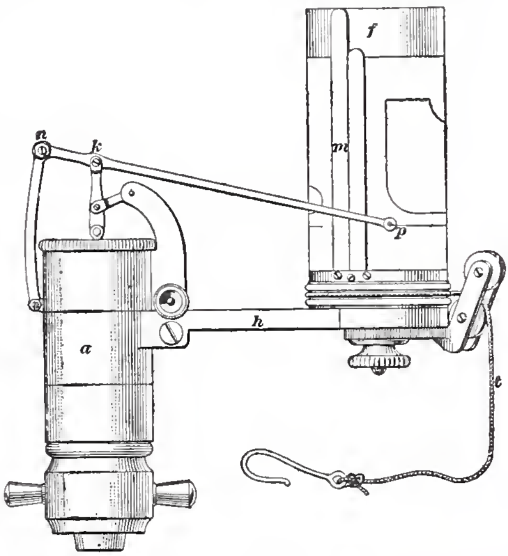


FIG. 34

58. Connecting Indicator to Steam Cylinder.—To attach the indicator to the engine, holes are drilled in the clearance spaces of the cylinder and tapped for a ½-inch

nipple, as shown at *h* and *c*, Fig. 35. It is preferable to have an indicator at each end of the cylinder, but if that is

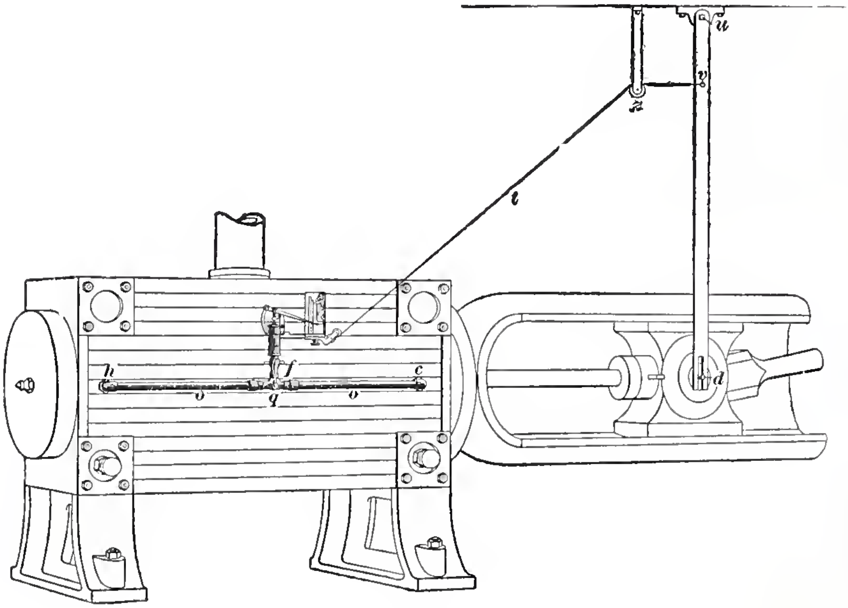


FIG. 35

not convenient, one indicator may be connected with both ends of the cylinder by means of a three-way cock, as shown in Fig. 35.

59. Reducing Motion.—The motion of the indicator drum *f*, Fig. 34, is nearly always taken from the crosshead, but since the stroke of the crosshead is longer than the circumference of the drum, it is necessary to have a device that will proportion the motion of the pencil moving on the card to the length of the stroke of the engine. Such a mechanism is called a **reducing motion.**

A common form of reducing motion, known as the slotted swinging lever, is shown in Fig. 35. A pin *d* in the crosshead moves in a slot in one end of the lever, the other end of which is pivoted at *u* to some stationary object. The cord *t* of the indicator is attached at *v*; the pin *u* should be directly over the pin *d* when the crosshead is at the center of its stroke.

Rule.—*To find the distance $u\,v$ between the pivot u and the point v where the cord is attached, multiply the length of the lever $u\,d$ by the desired length of the diagram, and divide the product by the length of stroke of the engine, all the dimensions being taken in inches.*

The reducing motion shown in Fig. 36 is easy to construct, and, if properly designed, gives very accurate results. The rod $U\,W$, which is pivoted at U, is connected to the crosshead by a swinging link $W\,D$. The cord may be attached to the sector $E\,F$, at E, or it may be fastened to a pin at point V. The end W of the link $W\,D$ moves in an arc about point U as a center. The length $U\,W$ should be su h that the extreme positions of W above and below the line of motion of the point D are about equal. This requires that when the crosshead is in the middle of its stroke, the point W shall be in its lowest position.

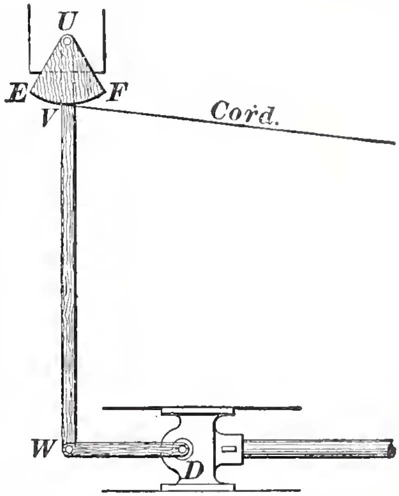


FIG. 36

EXAMPLE.—In Fig. 36 it is required to find the radius $U\,V$ of the arc $E\,F$ in order that the diagram may be $3\frac{1}{2}$ inches long, the stroke of the engine being 38 inches, and the length $U\,W$ being 5 feet 5 inches.

SOLUTION.—The length of the lever $U\,W$ is 5 ft. 5 in. = 65 in. Applying the rule,

$$U\,V = \frac{3.5 \times 65}{38} = 6 \text{ in., nearly. Ans.}$$

In addition to the reducing motions described, makers of indicators furnish pantographs, reducing wheels, and other devices for effecting the desired reduction. These devices are described in the catalogues of indicator manufacturers. For convenience, the cord should be in two parts: one attached to the reducing motion, and one to the indicator. The end of one cord is looped; the end of the other has a

hook attached. Instead of the cord a fine wire may be used, as the former has a tendency to stretch.

60. Directions for Taking Indicator Diagrams. Before attaching the indicator to the engine, blow steam through the pipe to remove dirt, and also see that the indicator is clean and in good working order. The piston should move freely. Oil the joints of the various levers and links and the piston with clock oil to avoid friction. Adjust the pencil so that it just touches the paper, and sharpen the point so that it makes a very fine light line. A heavy coarse line on a diagram indicates poor work.

Adjust the length of the cord so that the drum turns backwards and forwards without striking either of the stops at the end of the travel. When it touches one or the other of the stops, the cord is either too short or too long. If it touches both, the travel of the drum is too great, and the cord must be fastened to a point on the reducing motion having less travel.

Keep the drum moving only when taking diagrams. Unhook the cord before putting a paper or card on the drum. In putting on the card, see that it fits the drum without wrinkles, and fold back the projecting edges over the clips *m* (Fig. 34) so that they will not touch the pencil lever.

Before taking the diagram, turn on the steam a minute or so to warm the indicator; then, press the pencil lightly on the paper long enough to take a single diagram. Shut the cock *q* (Fig. 35) and again press the pencil to the paper. Since the indicator piston is th n only subjected to atmospheric pressure, the pencil will make a straight line called the **atmospheric line.** Disconnect the cord and remove the card. Write on the card the scale of the spring used, the size of the engine, its revolutions per minute, the boiler pressure, load, and the date, and any other desired particulars.

If one indicator is used for both ends, first open the three-way cock to admit steam from one end. Take the diagram

and open the cock to the other end, and take the diagram from that end. Then, shut off the steam entirely and take the atmospheric line.

61. Points and Lines of the Diagram. — Figs. 37 and 38 are indicator diagrams from the head and crank ends of an engine in which the approximate positions of the different points of the stroke are as follows:

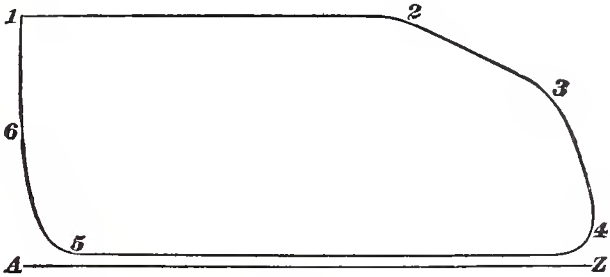


FIG. 37

1 is the beginning of the stroke; *2* is the point of cut-off; *3* is the point of release; *4* is the end of the stroke; *5* is the point of compression; *6* is the point of admission.

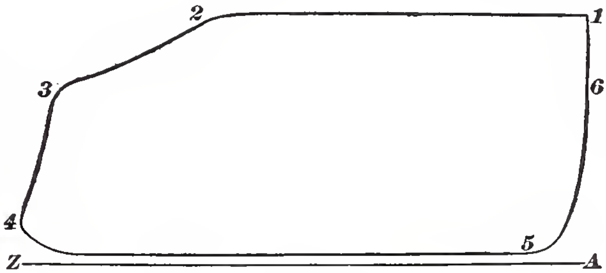


FIG. 38

The lines included between any two of these points have received special names, which are as follows:

6-1 is the admission line; *1-2* is the steam line; *2-3* is the expansion curve; *3-4-5* is the period of release; *4-5* is the back-pressure line; *5-6* is the compression curve; *A Z* is the atmospheric line.

It is sometimes desirable to have the vacuum line (line of no pressure) on the card also. The vacuum line may be

drawn below and parallel to the atmospheric line. The distance between them will be found in parts of an inch by the ratio $\frac{14.7}{\text{scale of spring}}$. Thus, if the scale of the indicator spring is 30, the vacuum line lies $\frac{14.7}{30} = .49$ inch below the atmospheric line.

62. If but one indicator is used, the two diagrams are taken on the same blank, as shown in Fig. 39. With the diagrams placed one over the other, as shown, it is possible to tell exactly what is taking place in the cylinder at any point

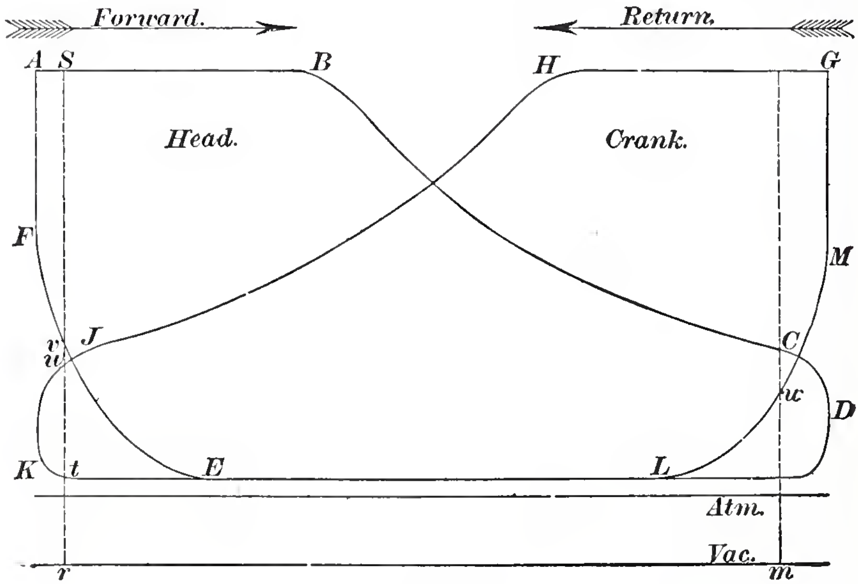


FIG. 39

of the stroke. On the forward stroke the pencil of the indicator describes the line $A\ B\ C\ D$ of the head diagram, if the cock is opened to the head end, or it describes the line $K\ L\ M$, if the cock is open to the crank end. The line $G\ H\ J\ K$ is described during the return stroke if the cock is opened to the crank end, or the line $D\ E\ F$ is described if the cock is opened to the head end.

In some indicators the cord is wound on the drum f in the reverse way from that shown in Fig. 34, so that the

head and crank diagrams change places from those shown in Fig. 39.

63. The **mean effective pressure,** usually written M. E. P., is the average pressure moving the piston forwards during one entire stroke, less the average pressures that resists its progress, that is, it is the difference in average pressures in the two ends of a steam cylinder during one stroke.

The mean effective pressure may be found in three ways: with the planimeter; by measuring the ordinates of an indicator diagram; and approximately by multiplying the total steam pressure by a constant that depends upon the cut-off.

64. *To Find M. E. P. With the Planimeter.*—The **planimeter** is an instrument for mechanically determining the area of an irregular figure, as for instance an indicator diagram. Some planimeters give the area in square inches,

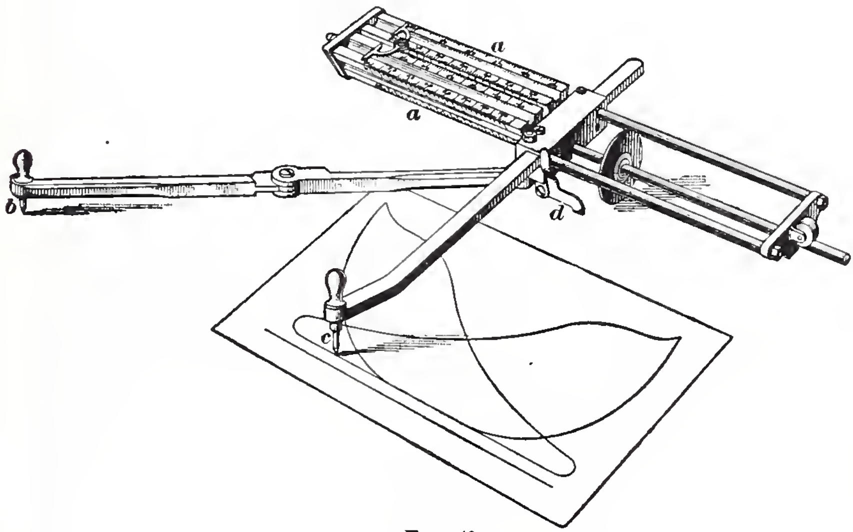


FIG. 40

and this area is then used in determining the mean effective pressure as follows: By dividing the area of the indicator diagram in square inches by the length of the diagram in inches, and multiplying by the scale of the spring.

EXAMPLE.—The area of the diagram is 4.2 square inches, and the length is 3.5 inches; a 40 spring being used, find the M. E. P.

SOLUTION.— $\frac{4.2}{3.5} \times 40 = 48$ lb. per sq. in., M. E. P. Ans.

Other more convenient planimeters give the readings directly for the mean effective pressure.

The Lippincott planimeter, shown in Fig. 40, consists of two boxwood scales *a*, supported on a metal frame. These scales contain 12 graduations, corresponding to 10, 12, 16, 20, 25, 30, 32, 40, 50, 60, 70, and 80 pounds indicator springs. These scales cover the range of all ordinary indicator springs. To use the planimeter, the point *b* is held firmly while the movable point *c* is moved about the outline of the figure whose area is desired. The instrument is set the length of the diagram to be measured by means of the small lever *d* and the outline is traced by the point *c*; the mean effective pressure can then be read directly from the scale corresponding to the indicator spring used in taking the diagram.

65. *To Find the M. E. P. by Ordinates.*—The following method of finding the M. E. P. is fairly rapid and accurate: Draw tangents to each end of the diagram perpendicular to the atmospheric line. Divide the horizontal distance between the tangents into 10 or more equal parts. Indicate by a dot on the diagram the center of each division, and draw lines through these dots parallel to the tangents, from the upper line to the lower line of the diagram. On a strip of paper mark off successively the lengths of these lines, so that the total length will represent the sum of all the lines. Divide this total length by the number of the lines used, and multiply the quotient by the scale of the spring. The result will be the M. E. P.

The diagrams are divided into 10 equal parts instead of some other number in order to assist in the work of calculation. All that is then necessary is to add the ordinates or vertical distances and shift the decimal point one place to the left to obtain the mean ordinate. This method saves the time required to divide by some inconvenient number, such as 14.

For example: The projection of the crank-end diagram on the atmospheric line in Fig. 41 is the distance *a z*, and it is

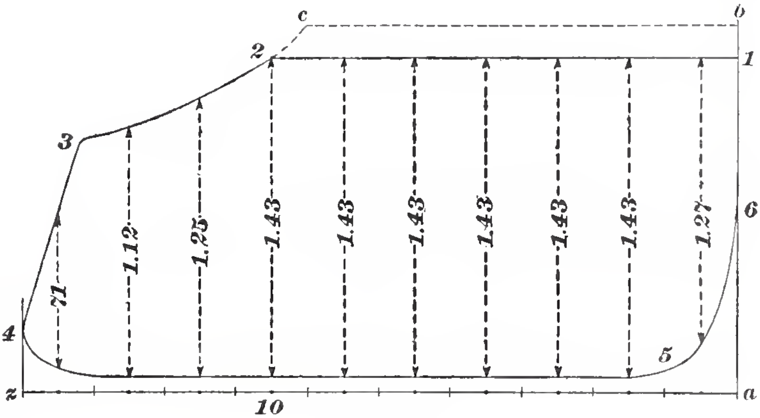


FIG. 41

divided in this case into 10 equal spaces. The length of each of the perpendicular lines drawn through the centers of these spaces is marked on the lines, and the sum of these lengths is 12.93 inches. If the scale of the spring used in obtaining the diagram was 40 pounds, then $\frac{12.93}{10} \times 40 = 51.72$ pounds per square inch is the M. E. P. of the crank-end diagram.

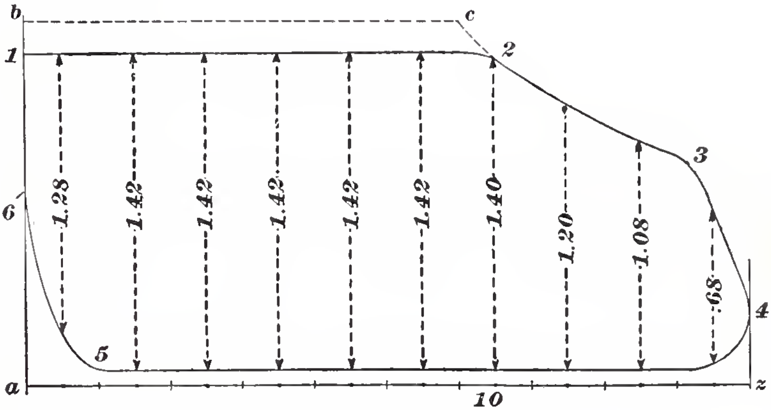


FIG. 42

The projection of the head-end diagram on the atmospheric line in Fig. 42 is the distance *a z*, and it is divided in

this case into 10 equal spaces. The length of each of the perpendicular lines drawn through the center of these spaces is marked on the lines, and the sum of these lengths is 12.74 inches. The scale of the spring is 40 pounds; therefore, $\frac{12.74}{10} \times 40 = 50.96$ pounds per square inch, or the M. E. P. of the head-end diagram.

Therefore, the M. E. P. in the cylinder during a complete revolution of the crank is $\frac{51.72 + 50.96}{2} = 51.34$ pounds per square inch.

Sometimes the expansion line of the diagram will fall below the back-pressure line, as shown in Fig. 43. In such a case, the sum of the ordinates included between C and D

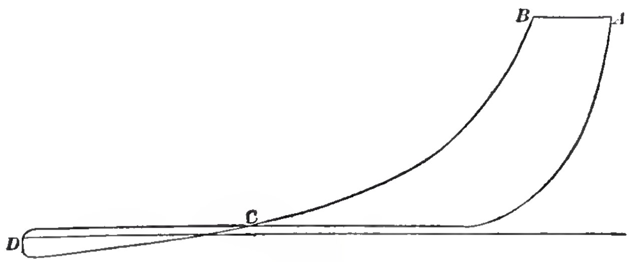


FIG. 43

must be subtracted from the sum of those included between C and A. The result divided by the total number of spaces will give the mean ordinate; multiplying this by the scale of the spring will give the M. E. P., as before.

66. *Approximate Determination of M. E. P.*—To approximately determine the M. E. P. of an engine when the point of apparent cut-off is known, and the boiler pressure, or the pressure per square inch in the boiler, from which the supply of steam is obtained, is given, and when an indicator diagram is not obtainable, use the following rule:

Rule.—*Add 14.7 to the gauge pressure and multiply the number opposite the fraction indicating the point of cut-off*

in Table II by this pressure. Subtract 17 from the product and multiply by .9. The result is the M. E. P. for good, simple non-condensing engines.

TABLE II

CONSTANTS FOR DETERMINING APPROXIMATE M. E. P.

Cut-off	Constant	Cut-off	Constant	Cut-off	Constant
$\frac{1}{6}$	.566	$\frac{3}{8}$	.771	$\frac{2}{3}$	.917
$\frac{1}{5}$	.603	.4	.789	.7	.926
$\frac{1}{4}$	.659	$\frac{1}{2}$	.847	$\frac{3}{4}$	.937
.3	.708	.6	.895	.8	.944
$\frac{1}{3}$	.743	$\frac{5}{8}$	.904	$\frac{7}{8}$	.951

If the engine is a simple condensing engine, subtract the pressure in the condenser instead of 17. The fraction indicating the point of cut-off is obtained by dividing the distance that the piston has traveled when the steam is cut off by the whole length of the stroke; i. e., it is the apparent cut-off. For a $\frac{2}{3}$ cut-off and 92 pounds gauge pressure in the boiler, the M. E. P. is $[(92 + 14.7) \times .917 - 17] \times .9 = 72.6$ pounds per square inch.

It is to be observed that this rule cannot be applied to a compound engine or any other engine in which the steam is expanded in successive stages in several cylinders.

EXAMPLE.—Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{1}{2}$ stroke and making 240 revolutions per minute. The boiler pressure is 80 pounds gauge.

SOLUTION.— $80 + 14.7 = 94.7$. The constant for $\frac{1}{2}$ cut-off is .847, and .847 × boiler pressure $= .847 \times 94.7 = 80.21$. M. E. P. $= (80.21 - 17) \times .9 = 56.89$ lb. per sq. in. Ans.

HORSEPOWER OF AN ENGINE

67. Indicated Horsepower.—Work is the product of force into the distance through which it moves. In the case of the engine cylinder, the total force is the M. E. P.

per square inch multiplied by the area of the piston; and the distance the piston moves in one minute is the number of strokes per minute multiplied by the length of the stroke.

Rule.—*To find the indicated horsepower developed by an engine, multiply together the M. E. P. per square inch, the area of piston in square inches, the length of stroke in feet, and the number of strokes per minute. This gives the work per minute in foot-pounds. Divide the product by 33,000; the result will be the indicated horsepower of the engine.*

If I. H. P = indicated horsepower of engine;
P = M. E. P. in pounds per square inch;
A = area of piston in square inches;
L = length of stroke in feet;
N = number of strokes per minute.

The above rule may be expressed thus:

$$\text{I. H. P.} = \frac{P\,L\,A\,N}{33{,}000}$$

The number of strokes which an engine makes per minute is twice the number of revolutions per minute. For example, if an engine runs at a speed of 210 revolutions per minute, it makes 420 strokes per minute. A few types of engines, however, are single-acting; that is, the steam only acts on one side of the piston. Such are the Westinghouse, the Willans, and others. In such engines, only one stroke per revolution does work, and, consequently, the number of strokes per minute to be used in the above rule is the same as the number of revolutions per minute.

Example.—The diameter of an engine piston is 10 inches, and the length of its stroke 15 inches. It makes 250 revolutions per minute, with an M. E. P. of 40 pounds per square inch. What is the horsepower?

Solution.—As it is not stated whether the engine is single-acting or double-acting, assume that it is double-acting. Then, the number of strokes is $250 \times 2 = 500$ per minute. Substituting the data in

$$\text{I. H. P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{40 \times \frac{15}{12} \times (10^2 \times .7854) \times 500}{33{,}000} = 59.5 \text{ H. P. Ans.}$$

In practice most engineers have pocketbooks containing tables of areas which they use, and to further shorten calculations, they generally take one decimal and even drop that if it be small. To illustrate, the table area of a 20-inch diameter circle is 314.16, but 314 is near enough for approximate calculations. Again, the table area of a 30-inch diameter circle is 706.86, and it may be taken at 707. In this way the preceding example would be figured 250 revolutions $\times$ 2.5 feet per revolution = 625 feet piston travel per minute. The area from a table is 78.5, and by its use the I. H. P. $= \dfrac{40 \times 625 \times 78.5}{33{,}000} = 59.469$, which may be approximately stated as 59.5 I. H. P.

68. Piston speed is the total distance traveled in feet by a piston in 1 minute. It is usual to take the length of the stroke (l) in inches. Then, to find the piston speed (S), multiply the length of stroke in inches (l) by the number of strokes (N) and divide by 12; that is, $S = \dfrac{l\,N}{12}$. But, since there are two strokes for each revolution, if R represents the number of revolutions per minute, $N = 2\,R$. Hence,

$$S = \frac{l\,N}{12} = \frac{l \times 2\,R}{12} = \frac{l\,R}{6}$$

Rule.—*To find the piston speed of an engine, multiply the stroke in inches by the number of revolutions per minute and divide the product by 6.*

EXAMPLE.—An engine with 52-inch stroke runs at a speed of 66 revolutions per minute. What is the piston speed ?

SOLUTION.— $S = \dfrac{l\,R}{6} = \dfrac{52 \times 66}{6} = 572$ ft. per min. Ans.

If the piston travel for one revolution is in even feet, we may shorten the calculation; thus, take the case of an engine with 42-inch stroke and 90 revolutions per minute; 42 inches equal $3\frac{1}{2}$ feet per stroke, or 7 feet per revolution; then the piston speed equals $7 \times 90 = 630$ feet per minute.

The piston speeds used in modern practice are about as follows:

	Feet Per Minute
Small stationary engines..........	300 to 600
Large stationary engines..........	600 to 1,000
Corliss engines....................	400 to 750
Locomotives........................	600 to 1,200

69. Frictional Horsepower and Net Horsepower. A part of the indicated horsepower is used in overcoming the friction of the moving parts of the engine. The remainder is available for doing work.

The power absorbed by the engine itself is termed **frictional horsepower.**

The power available for doing useful work is termed the **net,** or **actual, horsepower.**

The actual horsepower of any engine is found by first computing its I. H. P. from a set of indicator diagrams, taken when the engine is running under full load, and then subtracting from this the I. H. P. computed from a set of indicator diagrams, taken when the engine is running under no load, but making the same number of revolutions per minute as before. The horsepower developed by the engine in this latter case will only be sufficient to keep the working parts of the engine in motion at the same speed.

Example.—Indicator diagrams, taken from an engine when running under full load, showed an indicated horsepower of 242.7. With the same piston speed, and running under no load, the indicator diagrams showed an indicated horsepower of 38.2. Therefore, $242.7 - 38.2 = 204.5$, the actual horsepower of the engine.

70. The **mechanical efficiency** of an engine is the ratio of the actual horsepower to the indicated horsepower; or it is the percentage of the mechanical energy developed in the cylinder that is utilized in doing useful work. To find the efficiency of an engine, when the indicated and actual horsepowers are known:

Rule.—*Divide the actual horsepower by the indicated horsepower.*

EXAMPLE.—The indicator diagrams taken from an engine running under full load show the I. H. P. to be 238.5. The diagrams taken when the engine is running under no load show a horsepower of 39.7. (*a*) What is the net H. P. developed by the engine? (*b*) What is the efficiency of the engine?

SOLUTION.—(*a*) Net H. P. = I. H. P. − Friction H. P. = 238.5 − 39.7 = 198.8.

(*b*) By the rule, the efficiency is $\frac{\text{Net H. P.}}{\text{I. H. P.}} = \frac{198.8}{238.5} = 83.4\%$.

The mechanical efficiency of a good engine is from 75 to 90 per cent. of the mechanical energy developed in the cylinder.

EXAMPLES FOR PRACTICE

1. The area of an indicator diagram, as found by the planimeter, is 2.76 square inches. The length of the diagram is 2.4 inches, and the scale of the spring is 30. What is the M. E. P.?

Ans. $34\frac{1}{2}$ lb. per sq. in.

2. The indicator diagrams from a Corliss engine show a M. E. P. of 27.3 pounds per square inch. The engine has a 26″ × 48″ cylinder, and makes 68 revolutions per minute. Calculate the I. H. P. developed by the engine. Ans. 238.94 H. P., say 239 H. P.

3. Find the I. H. P. developed by an 8″ × 12″ engine, running at 260 revolutions per minute, the average M. E. P. being 32.61 pounds per square inch. Ans. 25.83 H. P., say 26 H. P.

4. (*a*) What is the piston speed of the engine of example 2? (*b*) Of the engine of example 3?

Ans. $\begin{cases} (a) & 544 \text{ ft.} \\ (b) & 520 \text{ ft.} \end{cases}$

5. The M. E. P. from the diagrams of an engine running under full load is 43.2 pounds per square inch. The M. E. P. from the diagrams when under no load is 5.7 pounds per square inch. The engine has a 16″ × 20″ cylinder, and makes 180 revolutions per minute. Find (*a*) the I. H. P. developed; (*b*) the frictional H. P.; (*c*) the net H. P.; (*d*) the mechanical efficiency of the engine.

Ans. $\begin{cases} (a) & 157.92 \text{ H. P.} \\ (b) & 20.84 \text{ H. P.} \\ (c) & 137.08 \text{ H. P.} \\ (d) & 86.8\%. \end{cases}$

6. An 18″ × 24″ engine, running at 150 revolutions per minute, cuts off steam at 18 inches of its stroke. Find (*a*) the probable M. E. P., and (*b*) the probable I. H. P. Boiler pressure, 70 pounds.

Ans. $\begin{cases} (a) & 56.127 \text{ lb. per sq. in.} \\ (b) & 259.68 \text{ H. P.} \end{cases}$

INTERPRETING INDICATOR DIAGRAMS

71. The indicator diagram not only furnishes a ready means of computing the horsepower developed by an engine, but it gives valuable information concerning the distribution of the steam. Any fault in the setting of the valves is at once revealed by an inspection of the diagram. The correction of such a fault may result in largely increased economy in the working of the engine.

The form of a good diagram depends largely on the type of the engine, style of valve, and speed. What would be considered a good diagram from a locomotive or from a high-speed automatic engine would be considered a very poor one if taken from a Corliss engine. In general, a diagram taken from an engine with releasing gear will be regular, and show but little compression, and the point of cut-off, release, and compression will be sharply marked. The diagram shown

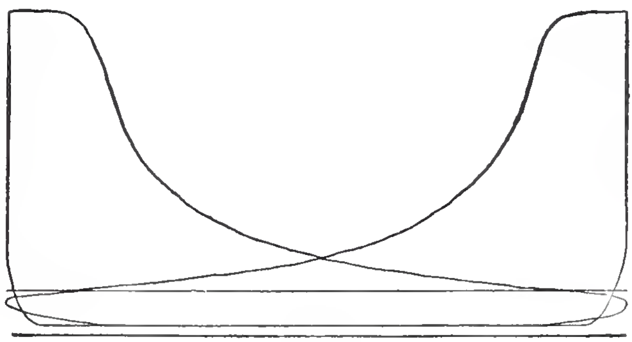


FIG. 44

in Fig. 44 is what may be expected from this type of engine when the valves are correctly set and in good working order. On the other hand, Fig. 45 shows the form of diagram that may be expected from an engine running at 250 to 300 revolutions per minute. On account of the high rotative speed, the lines are irregular, due to the inertia of the moving parts of the indicator. The compression is large, as it should be for engines running at a high speed.

It is readily seen how totally unlike are the diagrams shown in Figs. 44 and 45, yet each would be considered as representing good practice.

In the case of a hoisting engine or other intermittently running engine, a very instructive diagram may be obtained

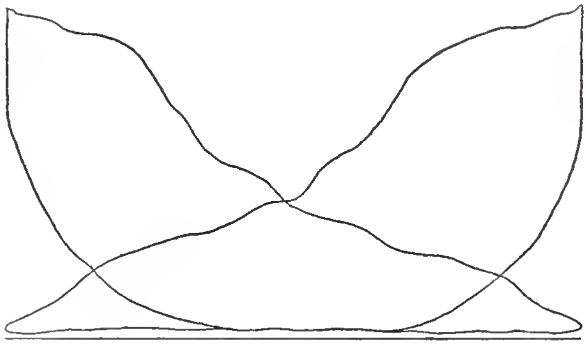


FIG. 45

by holding the pencil on the paper from start to finish during the hoisting of the load, and by the use of an indicator on each end of each cylinder, the power required at all points can be ascertained.

CONDENSERS

72. Condensation.—In non-condensing engines, that is, engines that are not supplied with a condenser, the steam is exhausted into the atmosphere, and therefore the exhaust steam must have at least the pressure of the atmosphere; in practice, the back pressure of steam in a non-condensing engine is scarcely ever less than 16 pounds above vacuum, and is oftener 17 pounds or more. The object in using a condenser is to create a partial vacuum and thus to save steam or to increase the power of the engine. In good condensing engines the back pressure is often as low as 2 pounds above vacuum.

73. Types of Condensers.—There are three types of condensers in general use: The *surface condenser*, the *jet condenser*, and the *gravity condenser*. In the surface condenser, the exhaust steam comes in contact with a large area of metallic surface that is kept cool by contact with cold water. In the jet condenser, the exhaust steam on entering the condenser comes in contact with a jet or spray of cold water. In either case, the entering steam is condensed to water, and since the water occupies vastly less space than the steam from which it is condensed, a partial vacuum is formed. If a sufficient amount of cold water were used, the steam on entering would instantly condense, and a practically perfect vacuum would be obtained were it not for the fact that the feedwater of the boiler always contains a small quantity of air, which passes with the exhaust steam into the condenser, and therefore partially destroys the vacuum. To get rid of this air, condensers are usually fitted with air pumps, which pump out both the air and the water into which the steam condenses.

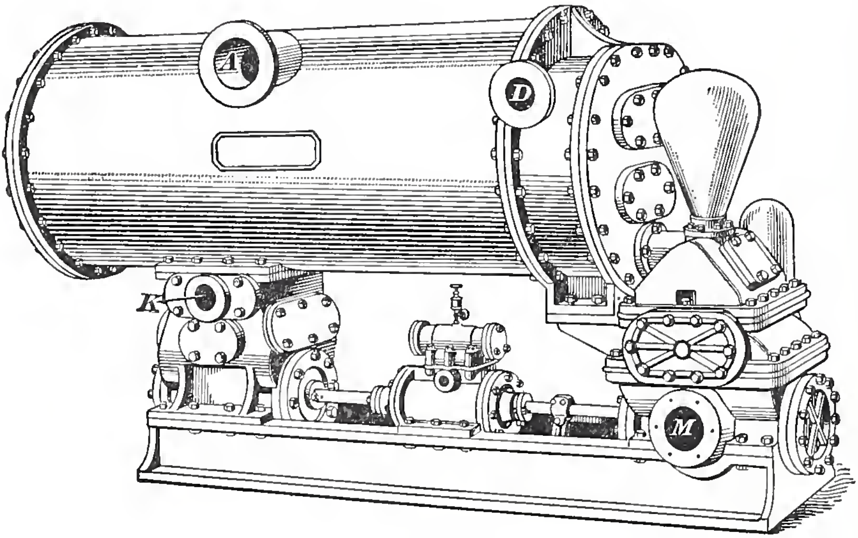


FIG. 46

74. Surface Condenser.— Fig. 46 is a perspective view of a surface condenser, and Fig. 47 sectional view.

The cold condensing water is drawn from some water supply through *M*, Fig. 46, and forced by the circulating pump *Q*, Fig. 47, into the inlet *C* of the condenser. From *C* the water is forced into the chamber *F*,

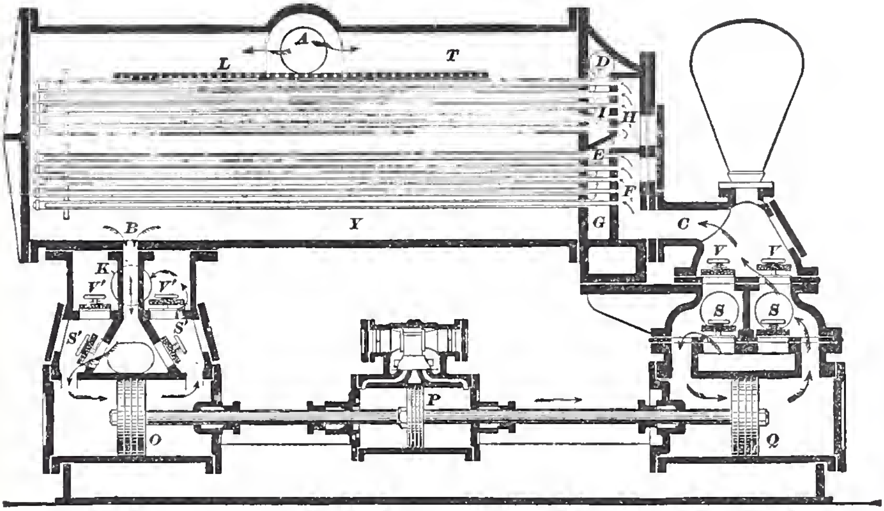


FIG. 47

and flows as indicated by the arrows through the inner tubes of the lower layer of double tubing to the left, when, having passed through their entire length, it returns through the space between the outside of the

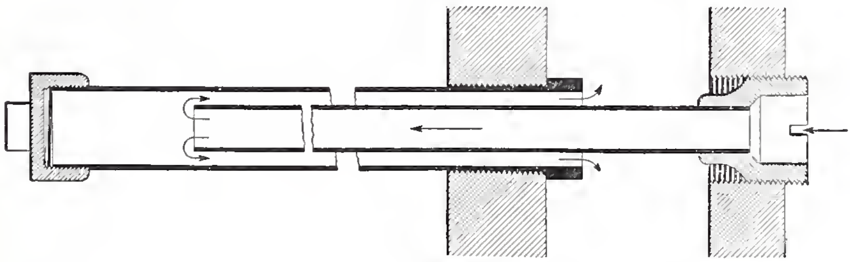


FIG. 48

inner and inside of the outer tubes into the chamber *G*. Fig. 48 shows more clearly the arrangement of this double tubing. From *G* (Fig. 47) it passes through *E* to *H*, and from *H* to *I* through the upper layer of double

tubing, as has already been explained. From I it is discharged through the nozzle D, carrying with it all the heat it has received by coming in contact with the two layers of double tubing.

The nozzle at A is connected with the exhaust pipe of the steam cylinder of an engine. The movement of the air-pump piston O draws the air in the condenser through

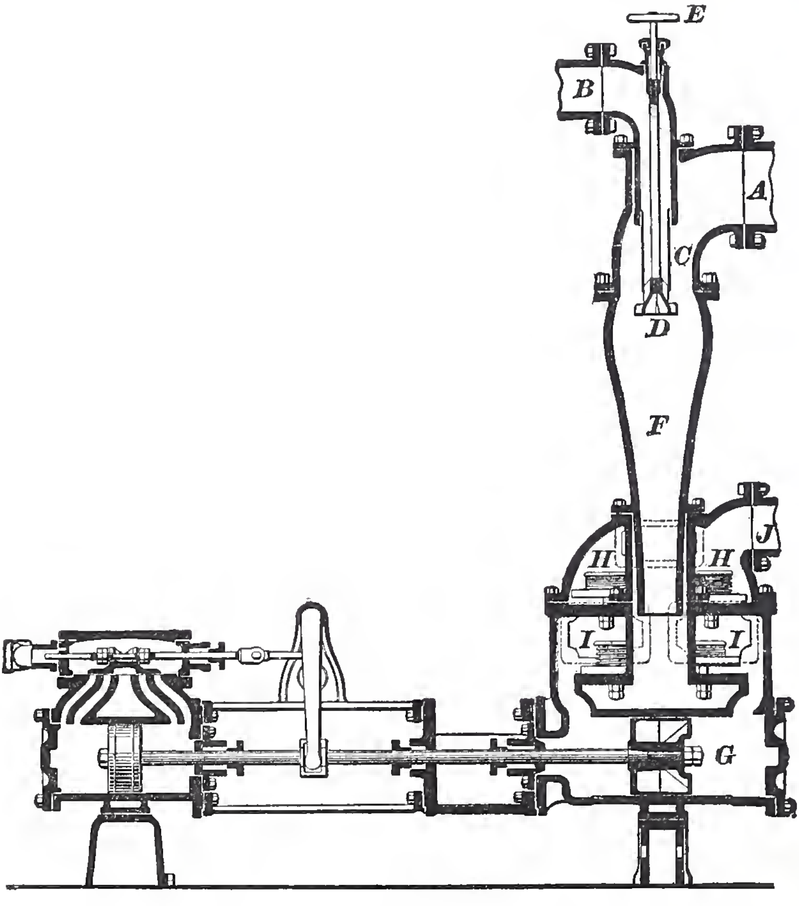


FIG. 49

the orifice B, and discharges it through the valves V' and nozzle K in a manner clearly indicated by the arrows. The valves S' and V' are opened and closed automatically by the pressure of the air beneath them and by the pressure of the air and springs above them. The air pump removes

the water of condensation and air that would otherwise destroy the vacuum.

As the exhaust steam enters the condenser cylinder through *A*, it first comes in contact with the perforated scattering plate *L* that protects the upper tubing from the damaging effect of direct contact with the exhaust steam. The steam then comes in contact with the tubes, through which the cold water is being pumped, and condenses. As soon as this occurs, the water of condensation collects at the bottom of the condenser cylinder and runs through *B* and into the air-pump cylinder, from which it is discharged while still heated and is made use of as boiler feedwater.

75. Jet Condensers.—Fig. 49 shows a section of an independent jet condenser. The cold water enters the

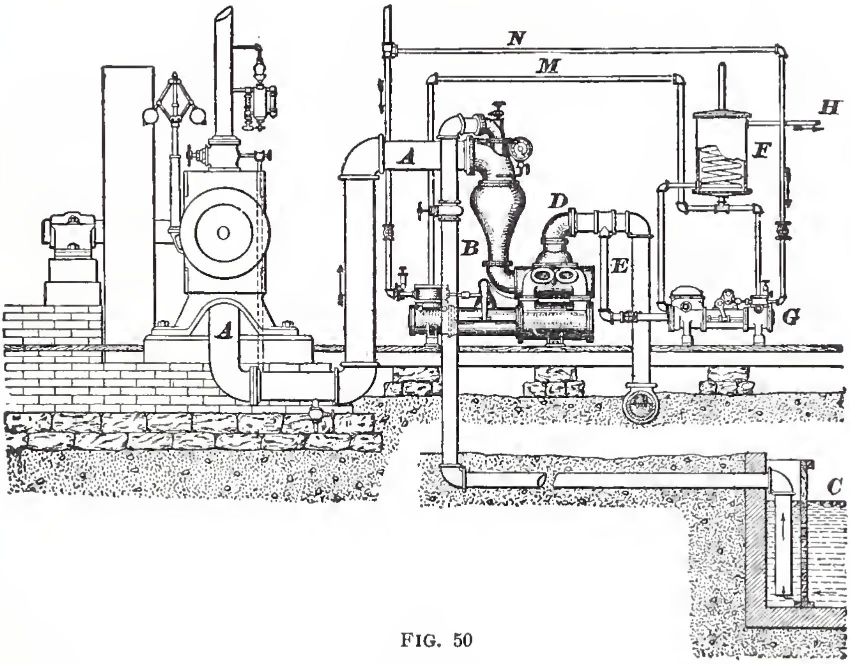


FIG. 50

condenser at *B*, passes down the spray pipe *C*, and is broken into a fine spray by the cone *D*. The exhaust steam in the meantime comes in at *A*, and mingling with the spray of

cold water, is rapidly condensed. The velocity of the entering steam is imparted to the water, and the whole mixture of steam, water, uncondensed vapor, and air is carried with a high velocity through the cone *F*, then through the valves *I* into the pump cylinder *G*, whence it is forced by the pump through the valves *H* into the discharge pipe *J*.

Fig. 50 shows a jet condenser in connection with an engine. The exhaust pipe *A* leads directly to the condenser. The injection pipe *B* draws water from the reservoir *C*. After the steam is condensed, the mixture of exhaust steam and injection water is discharged through *D*. A portion of this discharge, however, flows through *E* to the feed-pump *G*, which forces it through the coil in the heater *F* to the pipe *H* leading to the boiler. The exhaust from the two pumps is discharged into the feedwater heater through the pipe *M*. The water from the pipe *D* enters the feed-pump under a slight head, because the water is heated by the exhaust steam, and hot water cannot be raised by suction like cold water. *N* is a pipe leading from the boiler, and supplies steam for both pumps.

76. The **water required by a condenser** may be calculated by using the formula

$$W = \frac{H - t_3 + 32}{t_1 - t_2}$$

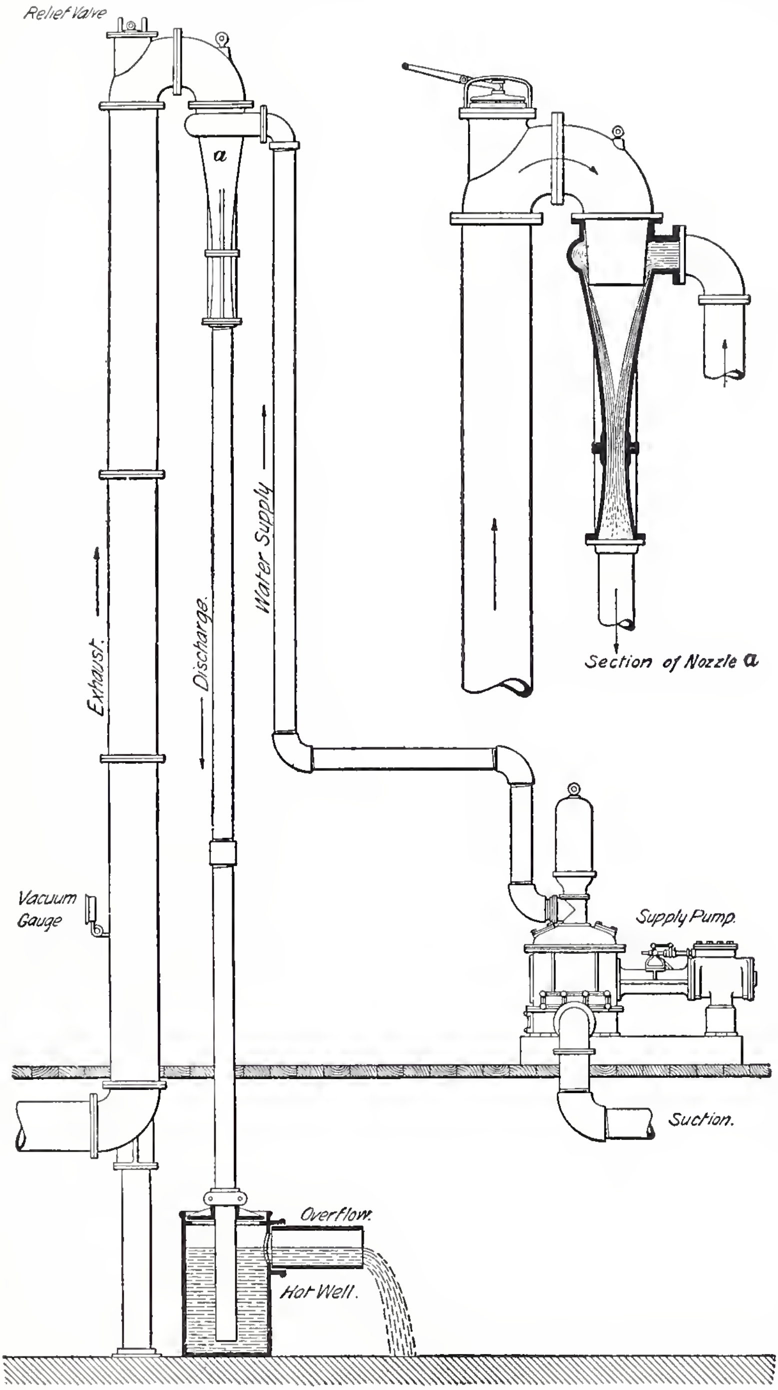
in which t_1 = the temperature of departing condensing water;

t_2 = the temperature of entering condensing water;

t_3 = the temperature of the condensed steam upon leaving the condenser;

H = total heat of vaporization of one pound of steam at the pressure of the exhaust; this may be obtained from a steam table;

W = weight of water required to condense a pound of steam.



Relief Valve
a
Exhaust.
Discharge.
Water Supply
Section of Nozzle a
Vacuum Gauge
Supply Pump.
Suction.
Overflow.
Hot Well.

FIG. 51

77. Gravity Condensers.—There are several forms of gravity condensers that require no air pump, depending on the principle that the weight of the atmosphere will not balance a column of water of over 32 feet (more or less) in height. Fig. 51 shows a gravity injector condenser. Exhaust steam enters through the exhaust pipe and nozzle *a*, and after passing through the nozzle comes in contact with water, which enters at the side openings about the nozzle and flows downwards in a circular ring, thus cooling and condensing the steam. The water of condensation is continuously discharged into the hot well, and as long as there is a column of water in the discharge pipe high enough to overcome the pressure of the atmosphere, there will be a continuous discharge through the overflow of the hot well. The water passes down through the contracted throat with such velocity as to carry with it all the air. A relief valve is provided that automatically opens and allows the exhaust steam to escape if the vacuum is lost.

78. Vacuum Gauge.—The degree of vacuum produced by a condenser is generally measured by a vacuum gauge that resembles a steam gauge in appearance and construction, but is graduated in inches instead of pounds for comparison with the barometer. The gauge is placed in any convenient location and connected by pipe to the condenser. If it were possible to obtain a perfect vacuum in the condenser, the gauge would give the same reading as a barometer at the same level, in other words, about 30 inches at sea level, or about 25 inches at an altitude of 5,000 feet above sea level. About 25 or 26 inches vacuum, as shown by gauge, at sea level is considered good practice in condensing engines. With no vacuum the pointer or hand will stand at zero. The reading of the vacuum gauge should be about 4 inches below the reading of the barometer to get the best results from the engine. A higher vacuum will result in a loss of heat from cold feedwater that will overbalance the gain from a higher vacuum, and it may also cause a high-speed engine to pound while passing the centers, through insufficient compression.

COMPOUND AND MULTIPLE-EXPANSION ENGINES

79. If an attempt is made to obtain a very high ratio of expansion in a single-cylinder engine, the gain due to the high ratio of expansion is lost by the condensation of the entering high-pressure steam, this loss being occasioned by the cooling of the cylinder walls during expansion and exhaust. Aside from the use of a condenser, the only available way to increase the amount of heat extracted from the steam by the engine is to raise the temperature and pressure of the entering steam. Following out this idea, steam pressures have been steadily increased from 8 to 10 pounds (above atmosphere), in the time of Watt, to 150 to 200 pounds per square inch, the pressures used in modern locomotives and marine engines.

Theoretically, it would be possible to make the efficiency of steam as high as desired by increasing indefinitely the pressure of the entering steam. Practically, this cannot be done since the steam pressure is limited by the strength of the boiler, and the engine cylinders cannot be made non-conductors of heat. When the fresh steam from the boiler enters the cylinder, it comes in contact with the walls that have just been reduced in temperature by the outgoing exhaust. Consequently, a portion of the steam immediately condenses, and gives up enough heat to reheat the walls. The steam thus condensed does no work. It is plain that by increasing the difference in temperature between the live and exhaust steam the loss due to cylinder condensation is largely increased. To illustrate this, if the absolute pressure of the steam passing into the condenser be 4 pounds, its temperature is about 153°. If the absolute pressure of the entering steam be 60 pounds, its temperature is about 293°. The fall in temperature is $293° - 153 = 140°$. Suppose, however, the entering steam had an absolute pressure of 200 pounds, its temperature would then be 382°, and the fall in temperature during the stroke would be $382° - 153° = 229°$. Now, it is plain that a great deal more of

the incoming steam must condense to raise the temperature of the cylinder walls back from 153° to 382° than to raise them from 153° to 293°. Hence, increasing the range of temperature increases the loss due to cylinder condensation.

80. Principle of Compounding.—To obtain the advantages of a high pressure, and at the same time avoid the loss due to cylinder condensation as much as possible, the steam may be allowed to expand successively in two or more cylinders. The fall of temperature is thus divided between the two or more cylinders, and consequently the loss from condensation in both or all of them is made considerably less than it would be if the same fall of temperature were allowed to take place in one cylinder. When the expansion takes place in two cylinders, the engine is said to be **compound**; if the expansion takes place successively in three cylinders, the engine is said to be **triple-expansion,** and if in four cylinders, **quadruple-expansion.**

It is important to note, however, that the expressions compound, triple, quadruple, do not refer to the number of cylinders, but, rather, to the number of successive expansions the steam is subjected to. For example, take two engines, each with three cylinders. In the first engine, if the steam expands in the smallest of the three cylinders, then passes into the next larger and expands there, and finally undergoes a third expansion in the largest cylinder, the engine is a triple-expansion engine, because the steam has had three separate expansions. In the other engine, if the steam, after expanding in the smallest cylinder divides, and each half expands in one of the other cylinders, the steam undergoes but two separate expansions, and the engine is a compound, though it has three cylinders. Similarly, a triple-expansion engine may have four or even five cylinders.

81. Mechanical Advantages of Compounding.—So far we have considered only the saving effected in the cylinders of the engine, as it is there that the greatest difference is shown; but there are other advantages incidental to the

use of triple-expansion and compound engines. The pressures, and consequently the strains, in a triple-expansion or a compound engine are more evenly distributed throughout the stroke than they are in a simple engine; therefore, lighter working parts can be used with the same margin of safety and lighter flywheels to give the same regularity of speed. These lighter weights and more even strains mean less friction in the engine, and this in turn means a saving of coal.

TYPES OF COMPOUND ENGINE

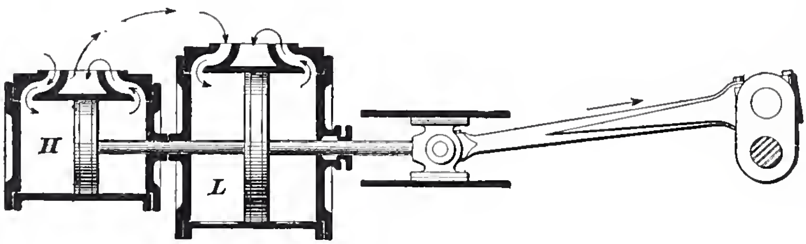
82. There are two general types of compound engines, the *tandem* and *cross-compound.*

The **tandem engine,** Fig. 52 (*a*), has its cylinders placed in line, the two pistons being attached to the same piston rod. *H* is the cylinder that first receives steam from the boiler; it is called the **high-pressure cylinder.** After the steam has expanded in *H*, it passes to the larger cylinder *L*, which is called the **low-pressure cylinder;** from here the steam is exhausted into the atmosphere or into a condenser.

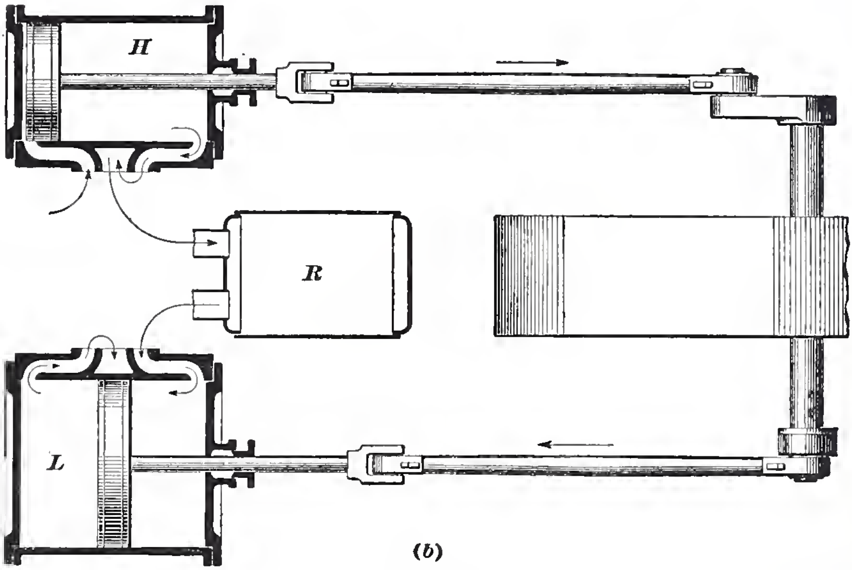
83. Fig. 52 (*b*) shows what is known as the **cross-compound engine,** where the cylinders are placed side by side and where the piston rods are attached to separate crossheads. The steam enters the high-pressure cylinder *H* from the boiler; exhausts into a separate vessel *R* called the **receiver;** from there it passes to the low-pressure cylinder *L*, and finally exhausts into the atmosphere or into a condenser. The cross-compound engine has two piston rods and two cranks; the cranks may be placed at any angle with each other, usually 90° apart.

The compound engine without a receiver may have one piston rod and crank, as shown in the tandem type, or it may have two piston rods and two cranks, the cylinders being placed side by side. In any compound engine, without a receiver, the two pistons must begin and end their

stroke at the same time, and the two cranks must be together or placed 180° apart. If both piston rods are attached to the same crosshead, the engine is called a **twin**



(*a*)



(*b*)

FIG. 52

compound. If any of these types of engines have a condenser, they are called **tandem, cross,** or **twin-compound condensing engines.** Without a condenser they are called **compound non-condensing engines.**

84. Size of Compound Engine.—In giving the size of a multiple-expansion engine, the stroke is always written last. Thus, a compound engine whose high-pressure cylinder is 11 inches in diameter, low-pressure cylinder 20 inches in

diameter, and stroke 15 inches, would be expressed as a 11″ and 20″ × 15″ compound. In the same manner, a 14″, 22″, and 34″ × 42″ triple-expansion engine would indicate that the diameters of the cylinders were 14, 22, and 34 inches, and that they had a common stroke of 42 inches.

85. The **ratio of expansion** of a compound or triple-expansion engine is the ratio between the volume of steam exhausted in the atmosphere or into the condenser per stroke and the volume of steam in the high-pressure cylinder at the point of cut-off.

Let e = ratio of expansion in high-pressure cylinder;
E = total ratio of expansion;
v = volume of cylinder receiving steam from the boiler;
V = volume of cylinder exhausting into atmosphere or condenser.

Then,
$$E = \frac{eV}{v}$$

Rule.—*The total ratio of expansion, or, as it is usually expressed, the number of expansions, is equal to the ratio of expansion of the small cylinder multiplied by the ratio between the volumes of the last and first cylinders.*

EXAMPLE.—In a compound engine, the volume of the low-pressure cylinder is 3.2 times that of the high-pressure cylinder, and the number of expansions in the latter is 2.4. What is the total ratio of expansion?

SOLUTION.— $E = \frac{eV}{v}$; but $e = 2.4$, and $\frac{V}{v} = 3.2$. Hence, $E = 2.4 \times 3.2 = 7.68$. Ans.

The total ratio of expansion varies from 6 to 12 for compound engines, and from 10 to 25 for triple-expansion engines.

EXAMPLES OF COMPOUND ENGINES

86. Tandem Compound.—An elevation of a tandem compound high-speed engine is shown in Fig. 53. In the illustration, b is the low-pressure cylinder, which in high-speed tandem compound engines is generally placed between

the high-pressure cylinder *a* and the crank. In large medium-speed engines the high-pressure cylinder is often placed nearest the crank-shaft, it being claimed that it is then easier to remove the pistons and to examine the cylinders in case of repairs, for if the low-pressure cylinder is placed as shown in Fig. 53, the high-pressure cylinder must be removed in order to get at the inside of the low-pressure cylinder. Steam is conducted to the high-pressure steam chest by the steam pipe *d*; after the steam has expanded in *a* it is discharged through the connecting

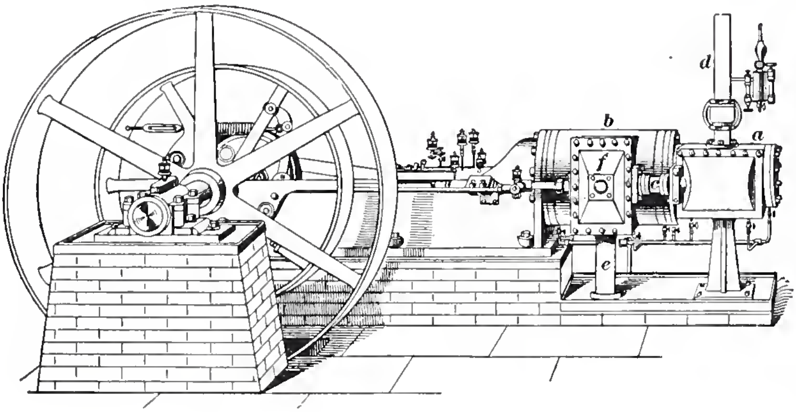


FIG. 53

pipe *c* into the steam chest *f* of the low-pressure cylinder *b*, and is finally exhausted into the condenser or atmosphere through the exhaust pipe *e*. In nearly all high-speed engines a shaft governor is used, which consists of one or more balls revolving about the shaft and drawn toward it by a spring. When the engine reaches the desired speed of revolution the centrifugal force overcomes the force of the spring and the ball moves outwards or away from the shaft, thus changing the position of the eccentric and causing an earlier cut-off. This tends to slow the engine, and the spring again draws the balls toward the shaft.

87. Horizontal Cross-Compound.—A perspective view of a cross-compound horizontal engine with Corliss valve

gear is given in Fig. 54. In the illustration, *a* is the high-pressure cylinder and *b* the low-pressure cylinder. The receiver between the two cylinders is placed beneath the floor, the pipe *c* leading to it from the high-pressure cylinder

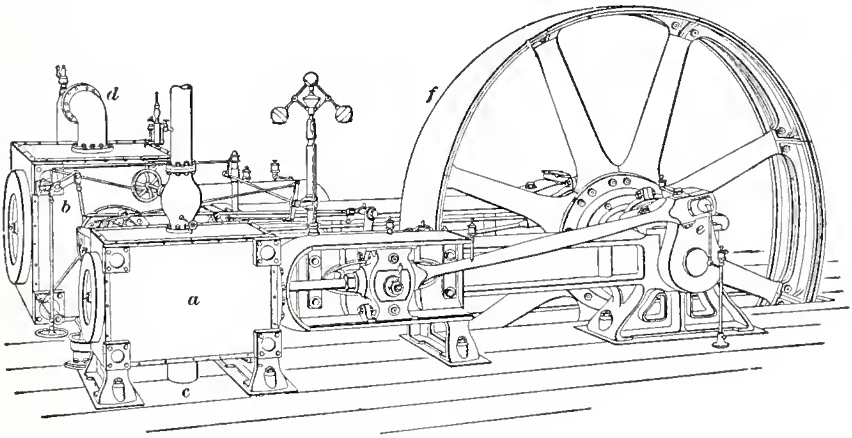


FIG. 54

and the pipe *d* leading from it to the low-pressure steam chest. The cranks are placed 90° apart; owing to the fact that the high-pressure crank *e* is on its upper quarter, the low-pressure crank is hidden by the frame. The flywheel *f* serves as a belt wheel and is placed between the cranks.

HORSEPOWER OF COMPOUND ENGINES

88. Calculating the Indicated Horsepower.—The indicated horsepower of a compound or triple-expansion engine is calculated from the indicator diagrams in exactly the same manner as with any simple engine, considering each cylinder as a simple engine and adding the horsepowers of the engines together. The indicator cards from a compound engine should be taken simultaneously from all cylinders, especially when the engine runs under a variable load, since otherwise an entirely wrong distribution of power may be shown.

89. Cylinder Volumes.—The average ratios of cylinder volumes of compound and triple-expansion engines with different steam pressures are given in the following Table:

TABLE III

AVERAGE RATIOS OF CYLINDER VOLUMES OF COMPOUND ENGINES

Initial Steam Pressure Gauge	High-Pressure Cylinder	Inter-mediate Cylinder	Low-Pressure Cylinder	Remarks
100	1		2.60	Non-condensing
100	1		3.60	Condensing
110	1		3.80	Condensing
120	1		4.00	Condensing
130	1		4.15	Condensing
140	1		4.30	Condensing
150	1		4.45	Condensing
160	1		4.60	Condensing
160	1	3.00	7.00	Condensing
185	1	3.33	7.80	Condensing

REVERSING GEARS

90. Stephenson Link Motion.—A reversing gear forms a part of every engine in which the direction of motion must be reversed. The most common form of reversing gear is the **Stephenson link motion,** which is shown in Fig. 55. Let O be the center of rotation of the crank C, and suppose the arrow to represent the direction of rotation of the engine. A is the eccentric and E the eccentric rod which control the valves when the engine is running under. B is the eccentric and F the eccentric rod which control the valves when the engine is running over, that is, in a direction opposite to that shown by the arrow in the figure. The

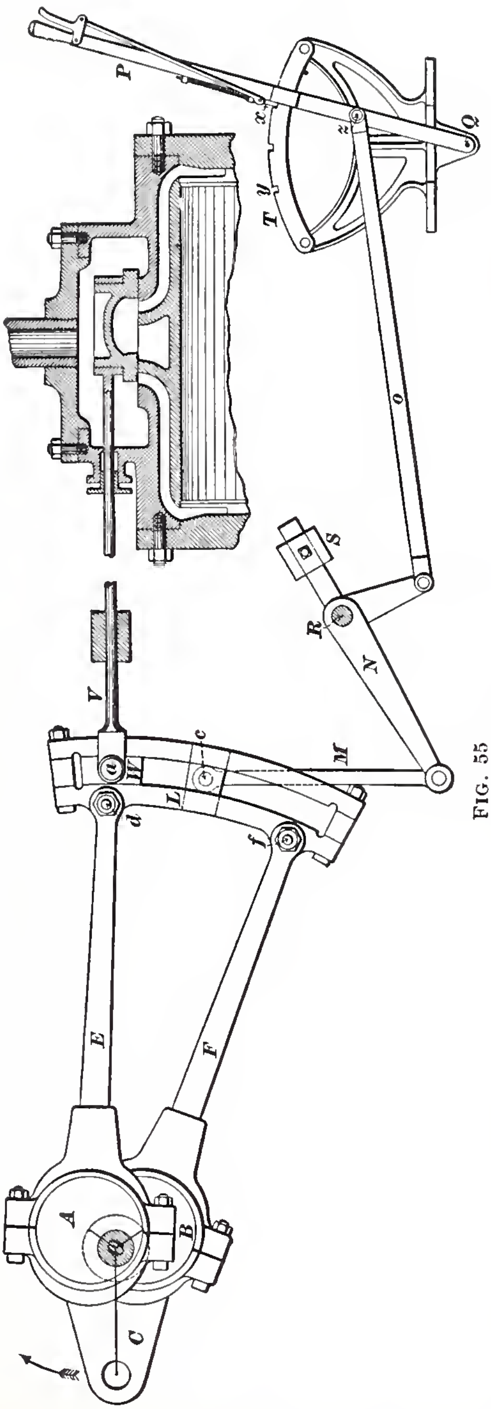


FIG. 55

eccentric A must be slightly more than a right angle in advance of the crank when it is directly connected, since it is to supply the means of operating the slide valve when the engine is rotating as shown. The parts as shown are in the positions required to cause the engine to run under. The eccentric B must be slightly more than a right angle behind the crank, when the engine is running under so that when the engine is reversed, and f takes the place of d, B may be in advance of the crank when the engine runs over. L is the reversing link; it has the form of an arc of a circle whose radius equals $O\,c$. The link block W forms the connection between the valve stem V and link L, and makes it possible for L to be moved through a distance fd. There is a joint between E and L at d, and another

between F and L at f. N is a bell-crank, and is rigidly connected to the tumbling shaft R, which is held in position by means of bearings. It is also jointed, as shown, to the lifting rod M and the reach rod o; M is connected to the center c of the link L, and the reach rod o is connected at z to the reversing lever P, which swings about Q as a center, and when moved is caught and held in its position by the spring latch x catching in the notches of the quadrant sector T. S is simply a counterbalance that balances the weight of the other parts about R as a center.

By a movement of the reversing lever P, through the length of the sector from the notch at x to the notch y, the point f is brought in line with a, and so changes the relative position of the slide valve (that is, of the port openings, etc.), that the engine can no longer rotate as indicated by the arrow, but must reverse its direction in consequence of the full steam pressure being brought to bear on the opposite side of the piston, as a result of this movement of the valve. Another important point, in connection with this link motion, is the fact that if the reversing lever is moved and secured so as to bring a between d and c, the valve travel will be reduced, and the admission port opening diminished, directly as the distance between a and c, in a's new position. When c reaches a, there will be no travel of the valve, and for points between c and f the valve travel will again increase directly as the distance between a and c increases.

COMPRESSED AIR IN COAL MINING

Serial 3031 Edition 1

COMPRESSED AIR

USES AND PROPERTIES

1. Uses.—Compressed air can sometimes be used to operate machines in gaseous coal mines, with greater safety than can any other form of power. Furthermore, for percussive mining machines and drills and for reciprocating pumps and hoists, compressed air has peculiar advantages. It can be carried from an outside plant into the mine more economically than can steam, and the exhaust from compressed-air machines is not objectionable, while exhaust steam in a mine is objectionable. Steam pipes must be covered to prevent condensation, and even then the loss is considerable. Compressed-air pipes, properly proportioned and installed are less subject to such objections. But in order to be sure of satisfactory results a compressed-air plant, including the distributing system, must be properly designed. The services, or at least the advice, of an experienced engineer are needed when the plant is installed.

2. Free Air.—Air under ordinary atmospheric conditions is spoken of as *free air,* and is generally considered to have a temperature of 60° F. and a pressure of 14.7 pounds per square inch at sea level. When close calculations are required, the exact atmospheric pressure at the particular time and place under consideration should be determined, for this pressure varies with atmospheric conditions and with the elevation, as shown in Table I.

3. Pressure.—The higher the elevation, the less is the atmospheric pressure, and at some point above the surface of the earth there is theoretically no atmospheric pressure. This theoretical point is called the *no pressure* or *absolute zero of pressure*, and pressure readings measured from it are considered as so many pounds absolute pressure per square inch. Thus the atmospheric pressure as measured by the barometer

TABLE I

PRESSURE AND BAROMETER READINGS AT VARIOUS ELEVATIONS

Elevation Above Sea Level Feet	Atmospheric Pressure Pounds per Square Inch	Barometer Reading Inches	Elevation Above Sea Level Feet	Atmospheric Pressure Pounds per Square Inch	Barometer Reading Inches
0	14.70	30.00	8,000	11.02	22.48
500	14.44	29.46	8,500	10.82	22.08
1,000	14.18	28.94	9,000	10.62	21.68
1,500	13.93	28.42	9,500	10.44	21.30
2,000	13.68	27.91	10,000	10.25	20.92
2,500	13.43	27.41	10,500	10.07	20.55
3,000	13.19	26.92	11,000	9.89	20.18
3,500	12.96	26.44	11,500	9.71	19.82
4,000	12.73	25.97	12,000	9.54	19.47
4,500	12.50	25.51	12,500	9.37	19.12
5,000	12.27	25.05	13,000	9.20	18.78
5,500	12.06	24.61	13,500	9.04	18.44
6,000	11.84	24.17	14,000	8.87	18.11
6,500	11.63	23.74	14,500	8.72	17.79
7,000	11.42	23.31	15,000	8.56	17.47
7,500	11.22	22.89			

is absolute pressure. In reality, at any pressure above absolute zero of pressure, air is compressed. Atmospheric pressure at sea level under normal conditions is 14.7 pounds more than zero pressure; that is, it is 14.7 pounds absolute. For convenience in calculating, atmospheric pressure is usually taken as 15 pounds per square inch, and most pressure gauges are made to read zero at this pressure. Therefore, when gauge pressure is

TABLE II

WEIGHT OF AIR AT VARIOUS PRESSURES AND TEMPERATURES

Temperature of Air. Deg. Fahr.	Gauge Pressure. Pounds																					
	0	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
	Weight in Pounds Per Cubic Foot																					
−20	.0900	.1205	.1515	.2125	.2744	.3360	.3970	.4580	.5190	.5800	.6410	.702	.7635	.825	.886	.948	1.010	1.165	1.318	1.465	1.625	1.930
−10	.0882	.1184	.1485	.2090	.2685	.3283	.3880	.4478	.5076	.5674	.6272	.687	.747	.807	.868	.928	.989	1.139	1.288	1.438	1.588	1.890
0	.0864	.1160	.1455	.2040	.2630	.3215	.3800	.4385	.4970	.5555	.6140	.672	.731	.790	.849	.908	.968	1.114	1.260	1.406	1.553	1.850
10	.0846	.1136	.1425	.1995	.2568	.3145	.3720	.4292	.4863	.5433	.6006	.658	.716	.774	.832	.889	.947	1.090	1.233	1.376	1.520	1.810
20	.0828	.1112	.1395	.1955	.2516	.3071	.3645	.4205	.4770	.5330	.5890	.645	.701	.757	.813	.869	.927	1.067	1.208	1.348	1.489	1.770
30	.0811	.1088	.1366	.1916	.2465	.3015	.3570	.4121	.4672	.5221	.5771	.632	.687	.742	.797	.852	.908	1.046	1.184	1.322	1.460	1.735
40	.0795	.1067	.1338	.1876	.2415	.2954	.3503	.4038	.4576	.5114	.5652	.619	.673	.727	.781	.835	.890	1.025	1.161	1.296	1.431	1.701
50	.0780	.1045	.1310	.1839	.2367	.2905	.3432	.3960	.4487	.5014	.5541	.607	.660	.713	.766	.819	.873	1.006	1.139	1.271	1.403	1.668
60	.0764	.1025	.1283	.1803	.2323	.2840	.3562	.3882	.4402	.4927	.5447	.596	.649	.700	.752	.804	.856	.988	1.116	1.245	1.376	1.636
70	.0750	.1005	.1260	.1770	.2280	.2791	.3302	.3808	.4316	.4824	.5332	.584	.635	.686	.737	.788	.839	.967	1.095	1.223	1.350	1.604
80	.0736	.0988	.1239	.1738	.2237	.2739	.3242	.3738	.4234	.4729	.5224	.572	.622	.673	.723	.774	.824	.949	1.047	1.199	1.325	1.573
90	.0723	.0970	.1218	.1707	.2195	.2688	.3182	.3670	.4154	.4639	.5122	.561	.611	.660	.709	.759	.809	.932	1.054	1.177	1.300	1.544
100	.0710	.0954	.1197	.1676	.2155	.2638	.3122	.3602	.4079	.4555	.5033	.551	.599	.648	.696	.745	.794	.914	1.035	1.155	1.276	1.517
110	.0698	.0937	.1176	.1645	.2115	.2593	.3070	.3542	.4011	.4481	.4950	.542	.589	.637	.685	.732	.780	.899	1.017	1.135	1.254	1.491
120	.0686	.0921	.1155	.1618	.2080	.2549	.3018	.3481	.3944	.4403	.4866	.533	.579	.626	.673	.720	.767	.884	1.001	1.118	1.234	1.465
130	.0674	.0905	.1135	.1590	.2045	.2505	.2966	.3446	.3924	.4296	.4770	.524	.570	.616	.662	.708	.754	.869	.984	1.099	1.214	1.440
140	.0663	.0889	.1115	.1565	.2015	.2465	.2915	.3364	.3813	.4262	.4711	.516	.561	.606	.651	.696	.742	.855	.968	1.081	1.194	1.416
150	.0652	.0874	.1096	.1541	.1985	.2425	.2865	.3308	.3751	.4193	.4636	.508	.552	.596	.640	.685	.730	.841	.953	1.064	1.175	1.392
175	.0626	.0840	.1054	.1482	.1910	.2335	.2755	.3181	.3607	.4033	.4450	.488	.531	.573	.658	.658	.701	.808	.914	1.021	1.128	1.337
200	.0603	.0809	.1014	.1427	.1840	.2248	.2655	.3054	.3473	.3882	.4291	.470	.511	.552	.592	.633	.674	.776	.879	.982	1.084	1.287
225	.0581	.0779	.0976	.1373	.1770	.2163	.2555	.2949	.3344	.3738	.4129	.452	.491	.531	.570	.609	.649	.747	.846	.944	1.043	1.240
250	.0560	.0751	.0941	.1323	.1705	.2085	.2466	.2845	.3223	.3602	.3981	.436	.474	.513	.551	.589	.627	.722	.817	.912	1.007	1.197
275	.0541	.0726	.0910	.1278	.1645	.2011	.2378	.2745	.3111	.3478	.3844	.421	.458	.494	.531	.568	.605	.697	.789	.881	.972	1.155
300	.0523	.0707	.0881	.1237	.1592	.1945	.2300	.2654	.3008	.3362	.3716	.407	.442	.478	.513	.549	.585	.673	.762	.852	.940	1.118
350	.0491	.0658	.0825	.1160	.1495	.1828	.2160	.2492	.2824	.3156	.3488	.382	.415	.449	.482	.516	.549	.632	.715	.799	.883	1.048
400	.0463	.0621	.0779	.1090	.1405	.1720	.2035	.2348	.2661	.2974	.3287	.360	.391	.423	.454	.486	.517	.674	.674	.753	.831	.987
450	.0437	.0586	.0735	.1033	.1330	.1628	.1925	.2220	.2515	.2810	.3105	.340	.369	.399	.429	.458	.488	.562	.637	.711	.786	.934
500	.0414	.0555	.0696	.0978	.1260	.1540	.1820	.2100	.2380	.2660	.2940	.322	.351	.379	.407	.435	.463	.534	.604	.675	.746	.885
550	.0394	.0528	.0661	.0930	.1198	.1464	.1730	.1996	.2262	.2528	.2794	.306	.333	.359	.386	.413	.440	.507	.573	.641	.709	.841
600	.0376	.0504	.0631	.0885	.1140	.1395	.1650	.1904	.2158	.2412	.2668	.292	.317	.343	.368	.393	.419	.483	.547	.611	.675	.801

given, the corresponding absolute pressure may be found approximately by adding 15 to the given pressure.

Compression may be expressed in pounds per square inch, as registered by a gauge; or it may be expressed in *atmospheres,* the number of atmospheres meaning the number of times atmospheric pressure and hence the number of expansions when used. For instance, air under 30 pounds gauge pressure would be under $30+15=45$ pounds absolute pressure, or a pressure of three atmospheres, since one atmosphere is taken as 15 pounds per square inch; the original volume of free air would be reduced to one-third of that volume, and it will expand to its original volume, or to three times its compressed volume on being liberated to free-air conditions.

4. Temperature.—The point at which there is absolutely no heat is called the *absolute-zero of temperature,* and experiments have been made which prove that this point is $-460°$ F., or 460 degrees below zero on a Fahrenheit thermometer. The *absolute temperature* corresponding to any Fahrenheit thermometer reading can therefore be found by adding 460 to the thermometer reading. Thus a temperature of 60 degrees above Fahrenheit zero represents an absolute temperature of $60+470=520$ degrees. A temperature of 12 degrees below Fahrenheit zero represents an absolute temperature of $-12+460$, or 448 degrees.

5. Weight of Air.—It has been determined that 1 cubic foot of air at an absolute temperature of 1° F., and at a pressure corresponding to a reading of 1 inch on the barometer, weighs 1.3273 pounds. The weight of 1 cubic foot of air at any other temperature and pressure may then be found by using the formula

$$W=\frac{1.3273\,P}{T}$$

in which W = required weight of 1 cubic foot of air, in pounds;
P = absolute pressure, in inches of mercury as shown by barometer;
T = absolute temperature of air, in degrees Fahrenheit.

EXAMPLE.—Find the weight of 1 cubic foot of air at a temperature of 60° F., when the barometer reads 30 inches.

SOLUTION.—The given values are $P=30$ and $T=60+460$; then by the formula,

$$W=\frac{1.3273\times 30}{60+460}=.0765 \text{ lb.}$$

Table II gives the weight of 1 cubic foot of air at various gauge pressures and temperatures, based on normal conditions at sea level, namely, gauge pressure 14.7 pounds per square inch, or 30 inches barometric reading.

PRESSURE AND TEMPERATURE LAWS

6. **Effect of Pressure on Volume of Air.**—Air is a mixture of gases, chiefly oxygen and nitrogen, and experiments have proved that when any gas is subjected to pressure its volume decreases as the absolute pressure increases. This variation of volume with pressure is so invariable that it can be stated as a law, which has been named *Boyle's Law,* or sometimes *Mariotte's Law,* as follows:

Boyle's or Mariotte's Law.—*The temperature remaining constant, the volume of a given weight of air varies inversely with the absolute pressure.*

This law may be expressed as a formula, thus:

$$pv=PV, \text{ or}$$

$$\frac{p}{P}=\frac{V}{v}$$

in which p = initial pressure;
P = final pressure;
v = initial volume;
V = final volume.

EXAMPLE.—If at an elevation of 5,000 feet above sea level 500 cubic inches of free air is compressed in the cylinder of an air compressor to a volume of 80 cubic inches, and the temperature remains constant, what is the gauge pressure of the compressed air?

SOLUTION.—The absolute pressure on the free air at 5,000 ft. elevation, as given in Table I, is 12.27 lb. per sq. in. The weight of a

given volume of free air does not change when it is compressed to a smaller volume. The values to be substituted in the formula are, therefore, $v=500$, $V=80$, $p=12.27$; and P, the higher absolute pressure, may be found by the formula

$$\frac{12.27}{P}=\frac{80}{500}, \text{ or } P=\frac{500\times 12.27}{80}=76.68 \text{ lb. per sq. in.}$$

The gauge pressure is, therefore, $76.68-12.27=64.41$ lb. per sq. in. Ans.

7. Effect of Temperature on Volume of Air.—Experiments have proved that change of temperature affects the volume of air or gases according to a fixed law, which has been named Gay-Lussac's Law, and is as follows:

Gay-Lussac's Law.—*The pressure remaining constant, the volume of a given weight of air varies directly with the absolute temperature.*

This law may be expressed as a formula, thus:

$$\frac{v}{V}=\frac{t}{T}$$

in which v = volume at lower temperature;
V = volume at higher temperature;
t = lower absolute temperature;
T = higher absolute temperature.

Example 1.—If 500 cubic feet of air having a temperature of 60° F. be heated to a temperature of 240° F. and allowed to expand freely, what will be its expanded volume?

Solution.—The two absolute temperatures are found by adding 460 to each of the given temperatures, making $t=60+460=520$ and $T=240+460=700$. The volume at the lower temperature $v=500$, and, by the formula,

$$\frac{500}{V}=\frac{520}{700}, \text{ or } V=\frac{500\times 700}{520}=678.08 \text{ cu. ft. Ans.}$$

Example 2.—If 800 cubic feet of air at a temperature of 225° F. is allowed to cool to 70° F., under constant pressure, what will be its volume?

Solution.—The two absolute pressures are 685 and 530. The volume at the higher temperature is 800, and, by the formula,

$$\frac{V}{800}=\frac{530}{685}, \text{ or } v=\frac{800\times 530}{685}=618.98 \text{ cu. ft. Ans.}$$

MOISTURE IN AIR

8. **Amount of Moisture.**—There is no such thing as dry atmospheric air. Whenever air is spoken of as "dry" it is only because it is drier than some other air. One of the most important and perpetual functions of the air in the atmosphere is the conveyance and distribution of moisture over the earth. Moisture in the air is usually in the form of invisible vapor, which may increase up to the *saturation point,* where the air can carry no more, and the vapor begins to condense into small particles of water visible as steam or fog. Rapid condensation causes rain. When the air is as full of vapor as it can be, the *dew point* is said to be reached and the humidity is said to be 100 per cent. The percentage of moisture in free air in different localities varies, as indicated by records of the weather bureau, of which the following are samples:

Galveston, Texas ...	85	Rapid City, S. Dak. .	60
New York City......	73	Salt Lake City, Utah.	53
Walla Walla, Wash. .	65	Yuma, Ariz.	42
El Paso, Texas.......	39		

Compressing air or changing its temperature changes the dew point, which is always passed when air is compressed to more than 6 atmospheres, of 75 pounds gauge. Compressed air may therefore be misty or wet, and the inner walls of compression chambers and pipes may be wet. In a long pipe line, a stream of water may form and pass along the bottom of the pipe with the air.

9. **Change of Saturation Point.**—Increasing the pressure on air lowers the saturation point and increasing the temperature raises this point. Compressing air raises its temperature, and its capacity to carry moisture is doubled with each rise of about 20° F. If free air were compressed to 6 atmospheres, 90 pounds absolute pressure, or 75 pounds gauge pressure, a common working pressure, and were kept cool during the compression, much of the moisture in it would be condensed. But such compression, even in water-cooled cylinders, causes tem-

peratures usually above 300° F., and the compressed air may, therefore, carry all the moisture with which it started.

The conditions under which compressed air will have the

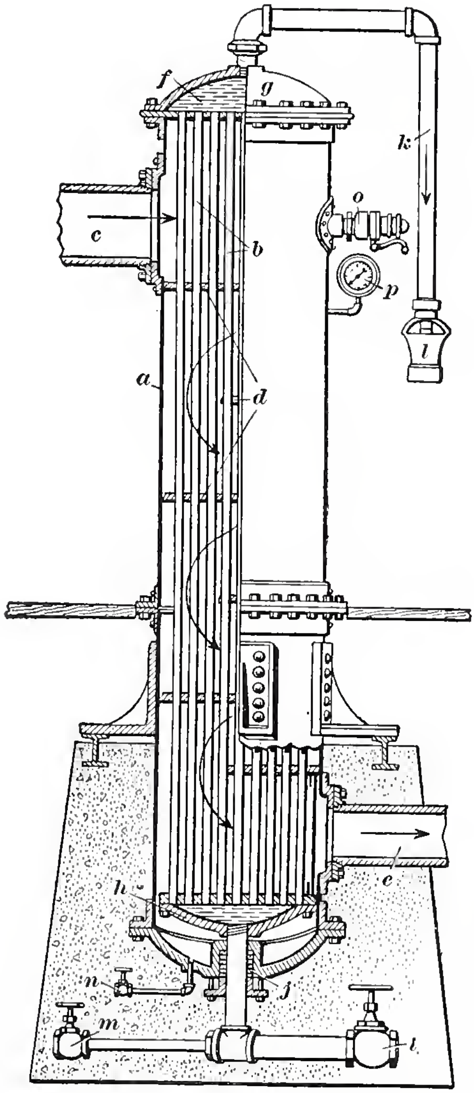


Fig. 1

lowest capacity for moisture are high pressure and low temperature. As the air leaves the compressor it is at its highest pressure and temperature. As it flows through the pipes its temperature falls, and condensation may take place, so that at

the point where work is to be done, the compressed air may carry very little moisture, if means are provided for draining condensation.

10. After-Cooling.—Water in air-pipe lines is objectionable; for it causes *water hammer,* such as is sometimes heard in steam-heating pipes, thus tending to cause leaky joints, and it accumulates in low places in pipe lines, thus reducing the air passage and causing loss of power. As much of the water as possible should, therefore, be removed from compressed air before it enters the pipe lines. This is often done by use of an after-cooler, one form of which is shown in Fig. 1.

This after-cooler consists of a vertical shell *a,* Fig. 1, in which the air circulates across a number of water tubes *b.* Air from the compressor enters the shell at *c,* is directed back and forth across the tubes by the baffle plates *d,* and flows through the connection *e* to the main pipe line. The upper ends of the water tubes are expanded into a plate or tube header *f* that is bolted between the shell and the cover plate *g* forming a water-tight compartment. The lower ends of the tubes are expanded into a tube header that is bolted to a false bottom *h* forming the lower water chamber. This construction allows for expansion and contraction of the water tubes caused by change of temperature. The supply of cooling water to the lower water chamber is controlled by a valve *i* and the feedpipe passes through a stuffingbox *j,* that forms a tight joint at this point but still allows for the expansion of the tubes. The circulation of cooling water is upwards through the tubes and out through the overflow pipe *k.* The funnel *l* forms a break in the overflow line and makes it possible for the attendant to gauge the temperature and the amount of cooling water used. At regular intervals the valve *i* should be closed and the valve *m* opened to drain or flush out the tubes, and any condensation that has collected in the bottom of the shell should be drained off through the valve *n.* The safety valve *o* prevents the pressure in the shell from exceeding the pressure for which the shell is designed. The pressure gauge *p* indicates at all times the exact pressure in the shell. The flow of air and water in opposite directions,

generally spoken of as counter-flow, as illustrated in this type of after-cooler, is very efficient, as the coldest air is affected by the coldest water, and consequently the final temperature of the air can be brought within about 10 or 20 degrees of the temperature of cooling water as it enters the tubes.

The use of an after-cooler helps to remove both moisture and lubricating oil from compressed air, thus rendering the air more suitable for use and less likely to injure any parts of the apparatus used to contain it. When compressed air saturated with moisture is used in a machine, ice forms around the exhaust ports; that is, the air is said to freeze. Lubricating oil suspended in compressed air injures rubber gaskets and rubber hose. Furthermore, an after-cooler lessens heat losses and con-

TABLE III

COOLING WATER REQUIRED PER 100 CUBIC FEET OF FREE AIR

	Cooling Effect Required	Gallons of Water at			
		60° F.	70° F.	80° F.	90° F.
1.	After-cooler or inter-cooler separate (80-100 lb. 2-stage compression)	2.5	3	3.5	4
2.	Inter-cooler and jackets in series (80-100 lb. 2-stage compression)	2.9	3.4	4.0	4.5
3.	After-coler for 80-100 lb. single stage compression	4.0	4.5	5.2	6.0
4.	Both low- and high-pressure jackets with water supply separate from inter-cooler (80-100 lb. 2-stage compression)	0.85	1.0	1.2	1.4
5.	Jacket for single-stage compression, 40-lb. air pressure	0.5	0.6	0.7	0.9
6.	Jacket for single-stage compression, 60-lb. air pressure	0.6	0.7	0.8	1.0
7.	Jacket for single-stage compression, 80-lb. air pressure	0.7	0.8	0.9	1.1
8.	Jacket for single-stage compression, 100-lb. air pressure	0.8	0.9	1.0	1.2

densation in air mains and results in less expansion of the air lines.

11. **Water Supply for Cooling.**—A moderate supply of cold water usually gives better cooling effect than does a larger supply of warmer water. In Table III is given the quantity of water at different temperatures that is required for various cooling effects in after-coolers and water-jackets of compressor cylinders. With these quantities, the temperatures of the cooled air and of the cooling water when it enters, will not usually differ more than 20° F. under ordinary working conditions.

In any case, enough water should be supplied to give proper cooling effect; and usually the results are best when the quantity is such that the water is discharged at a temperature of about 90° F. or 100° F. If a compressor is to stand idle in cold weather, all water spaces should be drained; freezing of water in pipes or water-jackets is likely to crack them. All water spaces should be washed out occasionally to remove sediment.

ADIABATIC AND ISOTHERMAL COMPRESSION AND EXPANSION

THEORETICAL COMPRESSION

12. Just as the temperature and pressure of air rises during compression, so both fall again when the air expands. Either compression or expansion is said to be *adiabatic,* meaning no heat transfer, when all of the heat remains in the air during the process. The word *isothermal,* meaning equal temperature, is used to refer to a process in which heat is withdrawn during compression, or restored during expansion, so as to keep the temperature uniform.

In practice, neither the true adiabatic nor the true isothermal condition is attained; some heat transfer must take place, for even with the best possible conditions some heat escapes during compression and cannot be recovered. Likewise it is not practical to prevent absorption of some heat during expansion. Neither is it practical to remove all of the heat of compression nor to restore all of the heat lost during expansion so as to maintain constant temperature during these actions. Actual

conditions for compression and expansion are between the two ideal conditions.

13. The conditions during compression may be represented by a curved line as in Fig. 2, in which the distance along the line AA' represents the volume of air, and the vertical dis-

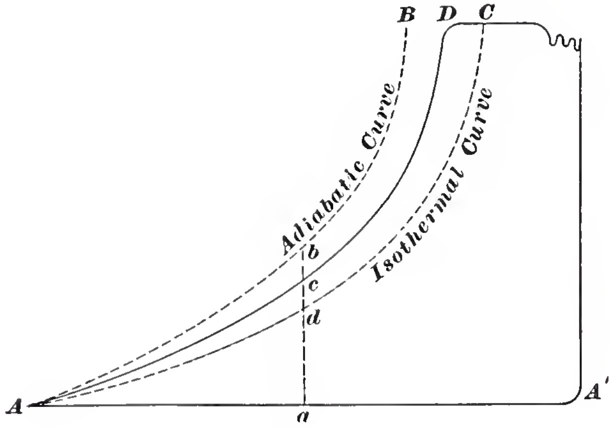


Fig. 2

tance from any point on this line to a curve represents the degree of compression at that point. For example, at a point a, the volume of compressed air is represented by the distance aA' and the compression by the vertical distance from the point a to one of the curves. If the compression were adiabatic, the compression would be proportional to the distance ab; if it were isothermal, the compression would be less, as represented by ad. The actual compression curve would be between the two theoretical curves as at AcD, its position relative to these two curves depending on the perfection of the cooling system. In practice, the actual curve is close to the adiabatic curve. The full lines in Fig. 2 show the form of an indicator diagram taken from an air-compressor cylinder.

The dotted lines AbB and AdC represent theoretical conditions and show clearly how rapidly the difference between adiabatic and isothermal compression, as represented by the distance db, increases as the compression increases.

REHEATING COMPRESSED AIR

14. Effect of Reheating.—Work is performed during the process of compressing air, and when the air expands again it performs work. If both compression and expansion could be done without any loss of heat, the work performed would be the same in both cases. But there is always some loss or transfer of heat with a corresponding loss of work during each

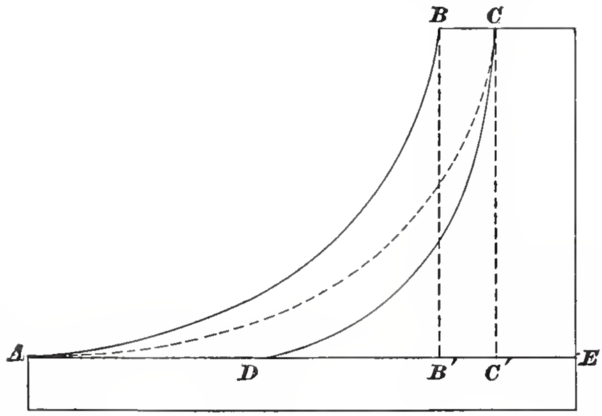


FIG. 3

process. The actual changes in temperature, pressure, volume and heat content of the air vary with working conditions, but a conception of the resulting loss of heat and the benefit of reheating can be gained by a study of the ideal curves shown in Fig. 3. These curves do not represent the exact conditions, but they serve to explain them. The gradual loss of heat through radiation that takes place during both compression and expansion and the loss due to cooling after compression are three separate actions, but the entire loss of heat may be considered to occur as a single action, taking place between the periods of compression and expansion. Compression and expansion then follow the adiabatic law.

The base line *AE*, Fig. 3, represents the length of piston stroke, and parts of this line represent proportional parts of the stroke. Vertical distances above the base line represent pressure. As the piston moves from *A* toward *E* the pressure rises, and if no heat could escape (adiabatic compression), the pressure

would rise according to the curve AB. If the heat of compression could be withdrawn to keep the temperature uniform (isothermal compression) the pressure would rise according to the curve AC. In the first case the piston would move a distance proportional to AB' to cause the maximum pressure, and the volume of compressed air would then be represented by the distance $B'E$. In the second case, isothermal compression, the piston would move a distance proportional to AC' to cause maximum pressure, and the volume of the compressed air would be represented by the distance $C'E$.

The distance $B'C'$, or BC, represents that part of the volume of the compressed air that is due to the heat of compression. If all this heat is removed in order to remove moisture, and the pressure is maintained constant during the removal, the volume will shrink to a volume corresponding to $C'E$, the same as that of air compressed isothermally, and the distance BC then represents the loss of heat.

If the air were then to expand adiabatically, from the point C the pressure would follow the curve CD and the volume would increase proportionately to the distance DC'. The loss of expansion due to cooling, and therefore the loss of work, is represented by AD. This loss can be regained by reheating the air to such a temperature that a curve representing the expansion will pass approximately through the point A. The heat added must be equal to or slightly greater than the heat lost by radiation and cooling. The expansion volume will then be represented by AC' so that the work done during expansion will be approximately equal to that of compression.

Reheating the compressed and cooled air also increases its ability to carry moisture and prevents the condensation of moisture in the pipes. Also, because of this reheating, the drop in temperature, due to expansion, is less likely to fall below the freezing point and there is less danger that ice will form at the point of exhaust.

15. Air Reheater.—By means of an air reheater, the temperature of compressed air may be raised about 250° F., thus increasing its volume and its capacity for doing work about

30 per cent. If a reheater is placed near the machines that use the air there will be little trouble from freezing air. The type of reheater shown in Fig. 4 has an inner fire-chamber surrounded by a cast-iron shell outside which is a sheet-iron shell with space for the air to be heated between the two shells. The outer shell is covered with non-conducting material to prevent escape of heat. Air enters through a pipe *b* at the top, passes in a thin layer between the two shells of the reheater and out at the bottom through a pipe *a* leading to the machines. Coal, coke, or other fuel may be used in the reheater.

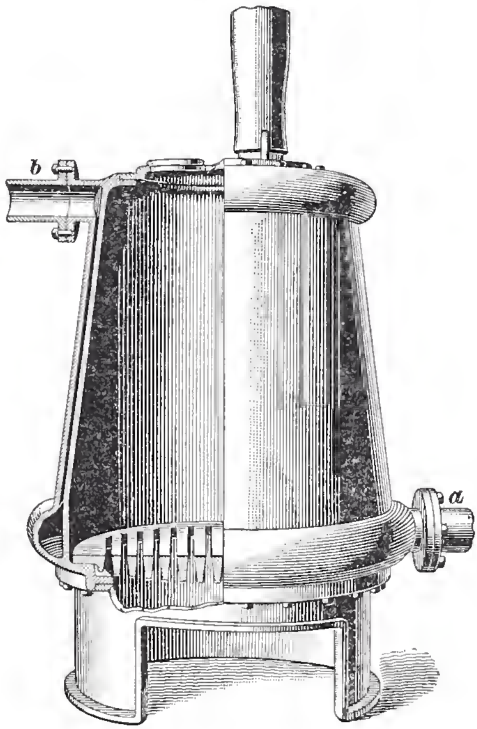


FIG. 4

The use of a reheater may be economical and advisable under some conditions. On the other hand, the cost of fuel and attendance may offset any possible gain. Reheaters are not permitted in coal mines or wherever an open fire is dangerous. The advice of experienced engineers, such as are employed by compressor manufacturers, should be sought before installing a reheater.

AIR COMPRESSORS

GENERAL DESCRIPTION

16. An air compressor is a machine having one or more cylinders, in each of which a piston is propelled to and fro. Suitable valves permit the entrance of free air behind the piston and the escape of compressed air at the required pressure from before the piston. The movements of the pistons may be caused by any available form of power, as steam, electricity, gas, or water.

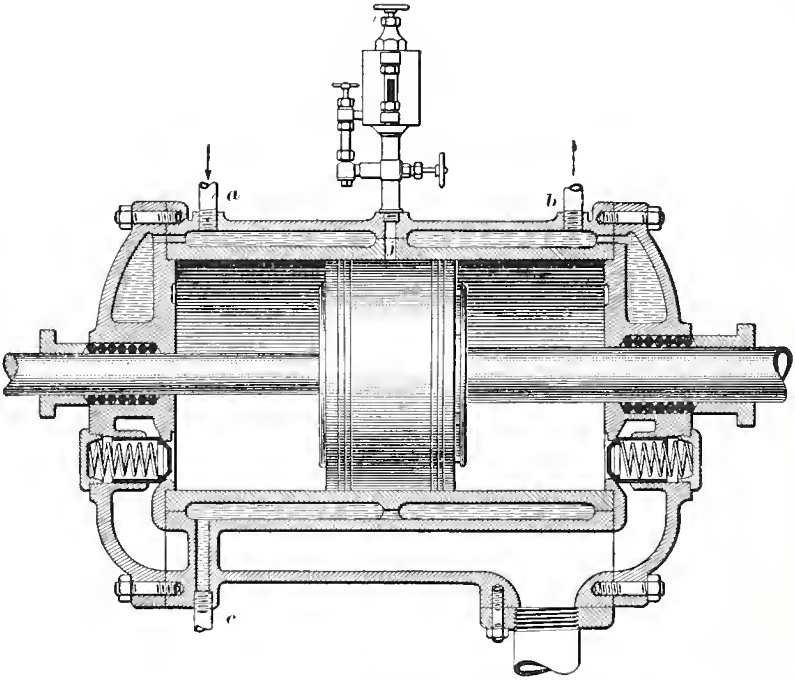


Fig. 5

Different methods of cooling the air as it is compressed have been tried, but the general practice is to circulate cooling water in a jacket around each cylinder. A common form of water-jacket is that in which the walls of the cylinder are made hollow, as shown in Fig. 5, cold water being made to circulate through the hollow spaces.

To secure proper circulation of the water the jacket is divided into two parts by a partition. Cold water enters through a

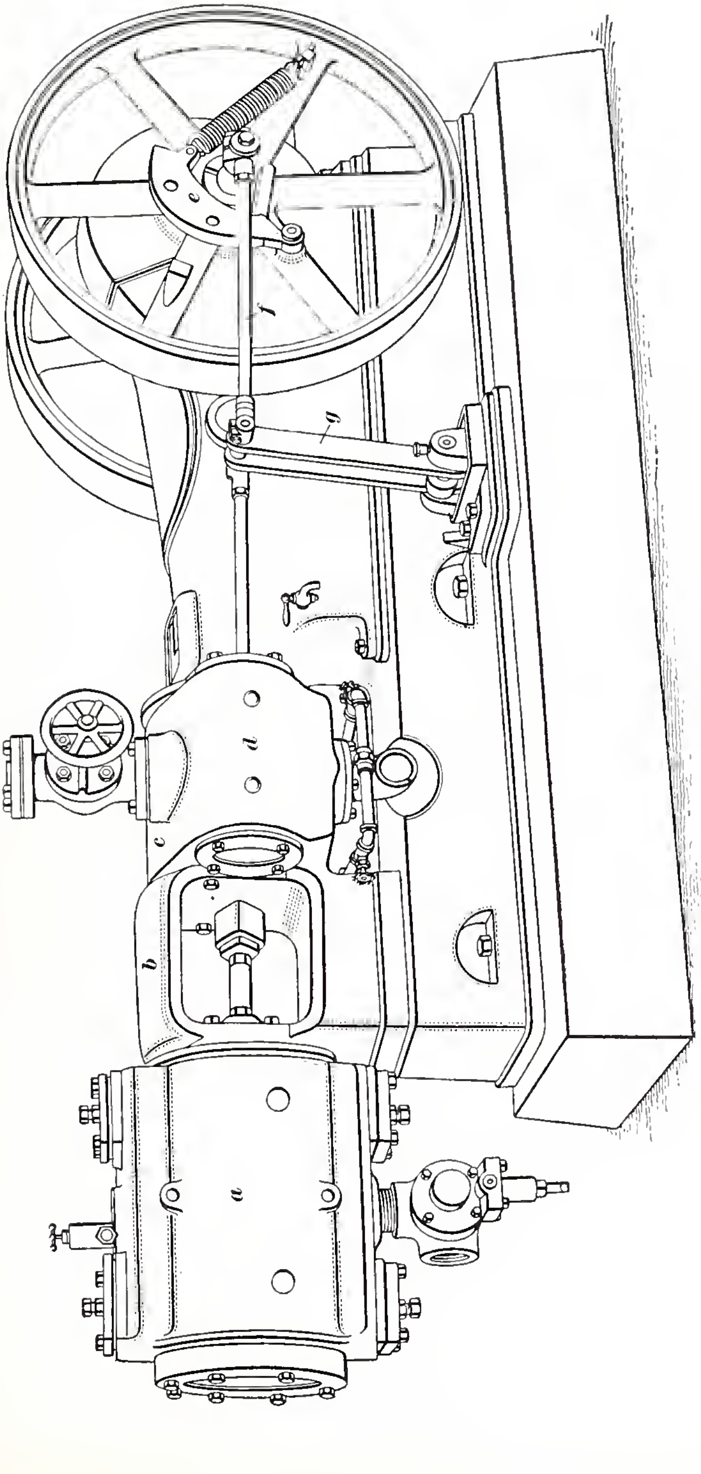


FIG. 6

pipe *a* connected with one of these parts and after circulating around that part it passes through an opening in the partition, and circulates around the other part, passing out through a pipe *b*. A pipe *c* provides for blowing out the jacket occasionally to clear it of sediment. The cooling water should be in contact with all of the surface of the cylinder that can be covered, and the piston speed should be moderate. At the very best, only part of the air in the cylinder comes in contact with the cooling walls. Adiabatic compression to 100 pounds gauge pressure in a dry compressor would cause a temperature of about 485° F., but the cooling arrangement should keep the maximum temperature of the compressed air below 350° F.

17. All water-jackets should have plugged holes, in order that they may be cleaned; also openings through which scale and mud can be scraped loose and washed out. It is good practice to keep the cooling surfaces clean, the frequency of washing depending on the purity of the cooling water used. If a jacket is not provided with openings for inspection and cleaning, at least two holes should be drilled, one near the top and one near the bottom. The air cylinder is usually made of cast iron, and in general form and arrangement is similar to the steam cylinder of a steam engine, except that it is water-jacketed.

CLASSES AND TYPES OF AIR COMPRESSORS

18. Classification.—Air compressors are classed as *vertical, horizontal, straight line,* and *duplex,* according to the position and arrangement of the cylinders. They are *single-acting* or *double-acting,* according to whether compression occurs with movement of the piston in one direction only or in both directions. Double-acting compressors are the more common. They are *single-stage* if all the compression occurs in one cylinder, and *multistage,* or *compound,* if the air while being compressed passes through more than one cylinder so that the process takes place in steps, or stages. Any compressor may be belt-driven or direct-connected with a source of power.

19. Single-Stage Compressor.—The steam-driven, single-stage compressor shown in Fig. 6 is designed to occupy mini-

mum floor space. The compressor cylinder *a* is attached by a spacer piece *b* to the steam cylinder *c* and the engine frame, so that both cylinders and the frame are on a common center line and thus form what is known as a *straight line compressor*. Economy in floor space is obtained by placing two small fly wheels at the end of the frame instead of between the cylinders, since by this arrangement a crosshead and connecting rod are eliminated with a corresponding reduction in over-all length. The steam chest *d* is located at the side of the steam cylinder.

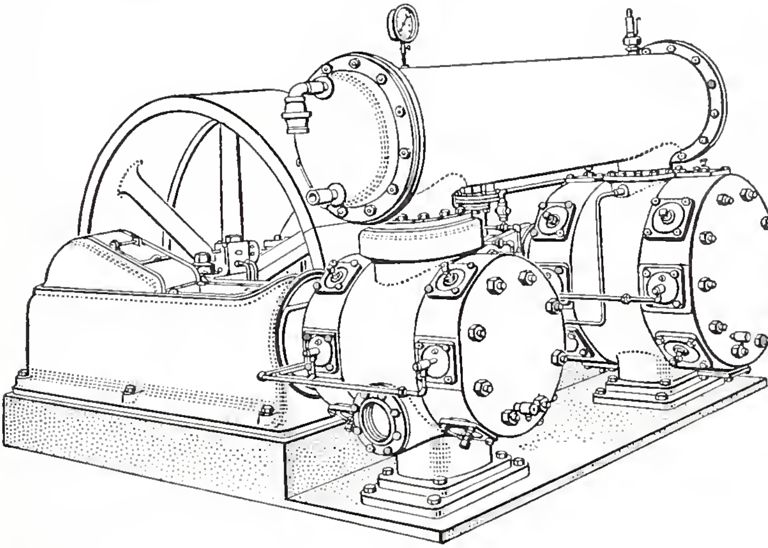


FIG. 7

The valve in this chest is operated by the governing mechanism *e* on one of the flywheels, through a reach rod *f* and rocker arm *g* attached to the compressor bed. The steam and compressor pistons are on a common piston rod, thus forming what is known as a direct-connected compressor.

20. Multistage Compressors.—A belt-driven two-stage, or duplex, compressor with an overhead inter-cooler is shown in Fig. 7. Free air enters the larger, or *low-pressure,* cylinder where it is compressed to a low pressure and discharged into the inter-cooler above. From the inter-cooler the air passes into the smaller, or *high-pressure*, cylinder where the compression is completed. The inter-cooler helps to remove heat from the air and thus to increase the efficiency of compression.

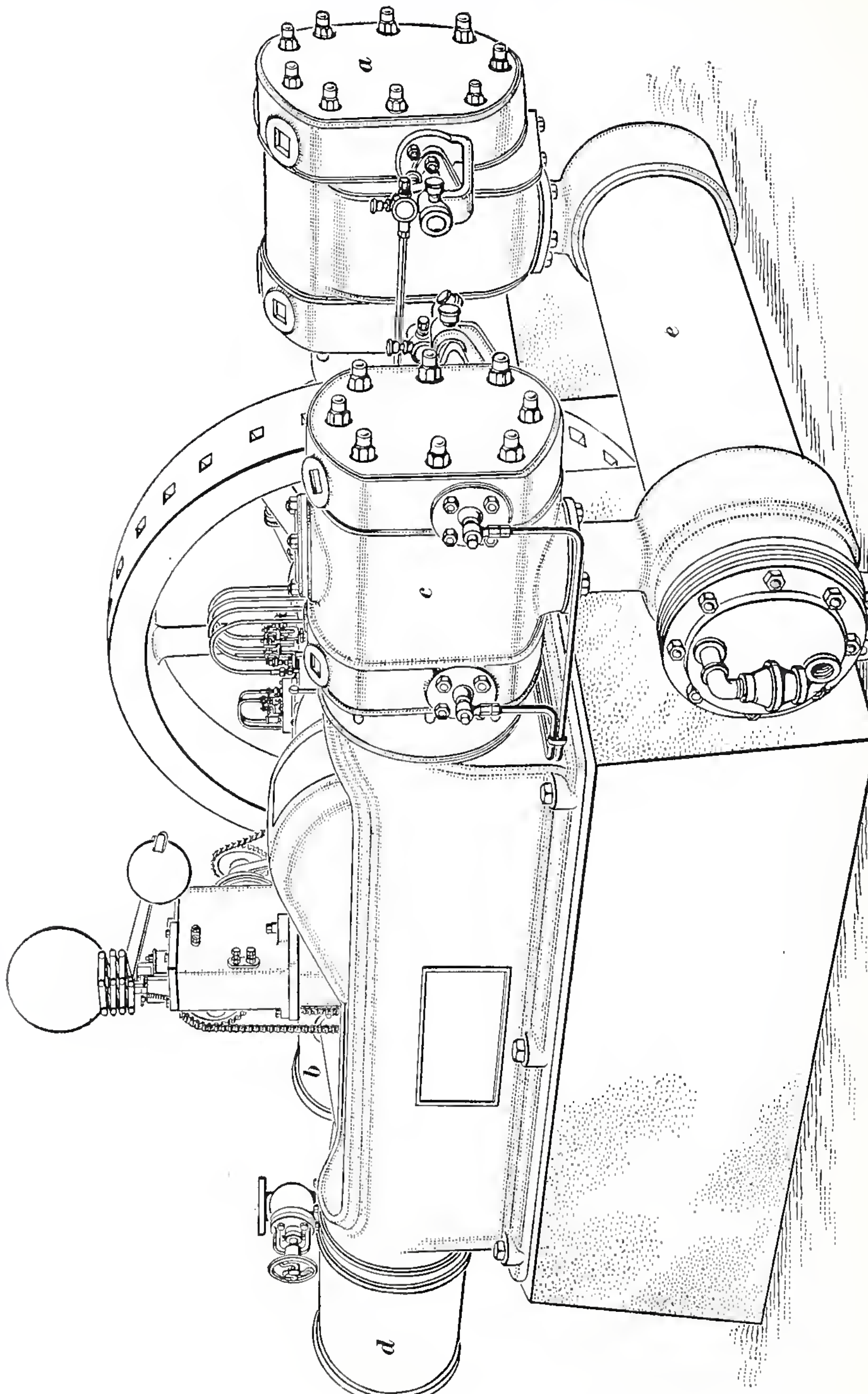


Fig. 8

A duplex compressor direct-driven by a compound engine is shown in Fig. 8. The low-pressure air cylinder *a* and the low-pressure steam cylinder *b* are in tandem, as are also the high-pressure air cylinder *c* and high-pressure steam cylinders *d*.

21. Compressors are also made for three- and four-stage operations with inter-coolers between stages. Pressures above 300 pounds gauge are usually required for compressed-air locomotives, and three-stage compressors are then advisable. Higher pressures up to 2,500 pounds are obtained with four-stage compressors, but such pressures are rarely used in coal mining. The high-pressure cylinder of three-stage compressors and the last two stages of four-stage compressors are often made single-acting. The reduction in the volume of the air as it is compressed and passed through the inter-coolers is so great that if double-acting compressor cylinders are used throughout, the higher-pressure pistons will have relatively small diameter, which might introduce mechanical difficulties. By using single-acting compressors for the higher pressures the size of the cylinders is more nearly uniform, and the pistons are of a convenient size for mechanical operation.

22. Advantages of Multistage Compression.—When air is compressed in one stage to a pressure of 80 or 100 pounds, the heat is so great as to interfere with the proper lubrication of cylinders and valves, as the oil forms a gummy deposit that obstructs the discharge ports and pipes. Compression in more than one stage helps to overcome this difficulty. Furthermore, high temperatures require constant watchfulness to prevent accumulations of oil deposits, carbon, and dust, on which the heat acts to form carbon monoxide gas, a little of which in the air is a deadly poison. Not only is this gas poisonous, but when mixed with air in right proportions (any proportion from 13 per cent. to 75 per cent. of gas), the mixture is explosive.

When air is compressed in two stages, the low-pressure cylinder raises the gauge pressure to about 30 pounds and the temperature to about 200° F. or possibly 260° F. This temperature depends on the effectiveness of the arrangements for cooling and on the temperature of the free air. The inter-coolers between

TABLE IV

COMPRESSION HORSEPOWER

For Adiabatic Compression of 1 Foot of Free Air at 14.7 Pounds Absolute Pressure, with 15 Per Cent. Allowance for Friction

Gauge Pressure Pounds per Square Inch	Single-Stage		Two-Stage		Three-Stage	
	M.E.P.	H.P.	M.E.P.	H.P.	M.E.P.	H.P.
5	5.12	.022				
10	9.44	.041				
15	13.17	.057				
20	16.44	.071				
25	19.47	.085				
30	22.21	.096				
35	24.72	.108				
40	27.05	.118				
45	29.21	.127				
50	31.31	.136	27.90	.123		
60	35.10	.153	31.30	.136		
70	38.59	.168	33.71	.147		
80	41.80	.182	36.15	.158		
90	44.71	.195	38.36	.167		
100	47.46	.207	40.48	.176	38.30	.167
110	50.09	.218	42.34	.185	....	....
120	52.53	.229	44.20	.193	....	....
130	54.87	.239	45.83	.200	....	....
140	57.08	.249	47.47	.207	....	....
150	59.18	.258	48.99	214	46.50	.202
160	61.80	.269	50.39	.219	....	....
170	64.00	.278	51.66	.225	....	....
180	65.80	.286	52.95	.231	....	....
190	67.70	.294	54.22	.236	....	....
200	69.50	.303	55.39	.241	52.00	.226
210	....	....	56.70	.247	....	....
220	....	....	57.70	.252	....	....
230	....	....	59.10	.257	....	....
240	....	....	60.10	.262	....	....
250	....	....	60.76	.264	56.60	.246
260	....	....	62.05	.270	....	....
270	....	....	62.90	.274	....	....
280	....	....	63.85	.278	....	....
290	....	....	64.75	.282	....	....
300	....	....	65.20	.283	60.70	.264
350	....	....	69.16	301	63.80	.277

TABLE IV—(*Continued*)

Gauge Pressure Pounds per Square Inch	Single-Stage		Two-Stage		Three-Stage	
	M.E.P.	H.P.	M.E.P.	H.P.	M.E.P.	H.P.
400			72.65	.317	66.90	.292
450			75.81	.329	69.40	.302
500			78.72	.342	71.70	.314
550					74.75	.326
600					76.90	.334
650					78.15	.340
700					79.85	.348
750					81.40	.355
800					83.25	.362
850					84.90	.369
900					86.00	.375
950					87.50	.381
1,000					88.80	.383
1,050					90.10	.391
1,100					91.10	.396
1,150					92.20	.401
1,200					93.15	.405
1,250					94.30	.411
1,300					95.30	.416
1,350					96.60	.421
1,400					97.30	.423
1,450					98.20	.426
1,500					98.80	.430
1,550					99.85	.434
1,600					100.80	.438

cylinders withdraw enough heat from the air so that its temperature on leaving each succeeding cylinder may be only a little higher than on leaving the low-pressure cylinder.

In comparison with compression in one stage, multistage compression has several other advantages. It requires less power for a given output, because the power requirement is more nearly uniform and the power unit can work more efficiently. The stresses on bearings, valves, and other parts are less in the multistage compressor, because the difference between the suction and the discharge pressures in each stage is less.

23. The saving in power made by multistage compression is shown in Table IV, which gives the mean effective pressures and horsepower consumption for compressing 1 cubic foot of free air at zero gauge pressure or 14.7 pounds absolute pressure per square inch to various pressures ranging from 5 pounds to 1,600 pounds gauge pressure per square inch. The values tabulated are obtained by adding 15 per cent. friction loss to the theoretical values found by the application of formulas. Single-stage compression is not given above 200 pounds per square inch, gauge pressure, but the table shows that economy may be obtained by compounding at gauge pressures as low as 50 pounds per square inch. A similar relation exists between two-stage compression, for which figures are given up to 500 pounds, and three-stage compression, which can be used economically for as low as 100 pounds gauge pressure per square inch. The chief factor in the choice of staging is the first cost of the compressor as compared to the quantity of air required.

TABLE V

VARIATION OF CAPACITY WITH TEMPERATURE OF FREE AIR

Initial Temperature Degrees F.	Relative Capacity	Initial Temperature Degrees F.	Relative Capacity
−20	1.18	70	.980
−10	1.155	80	.961
0	1.13	90	.944
10	1.104	100	.928
20	1.083	110	.912
30	1.061	120	.896
32	1.058	130	.880
40	1.040	140	.866
50	1.020	150	.852
60	1.000	160	.838

24. Capacity of Compressors.—The capacity of a compressor of any type depends on the temperature of the free air taken, because the air is more dense at low temperatures. The colder the free air the higher is the capacity of the compressor.

The relative capacities at different temperatures are shown in Table V. For example, if the capacity with free air at 60° F. is considered unity, the capacity with air at −20° F. is 1.18, and with air at 160° F. the capacity is only .838.

25. The capacity of a compressor is also affected by elevation, since the density of the air decreases with increased elevation just as in the case of increased temperature. Therefore, a compressor will not deliver as much air at a high elevation as it would at sea level with free air at the same temperature. This loss in delivery capacity may be considered from an

TABLE VI

VARIATION OF CAPACITY WITH ELEVATION

Altitude Feet	Barometric Pressure		Per Cent. of Sea Level Capacity	Loss of Capacity Per Cent.	Decrease in Required Power in Per Cent.
	Inches Mercury	Pounds per Square Inch			
0	30.00	14.75	100	0	0
1,000	28.88	14.20	97	3	1.8
2,000	27.80	13.67	93	7	3.5
3,000	26.76	13.16	90	10	5.2
4,000	25.76	12.67	87	13	6.9
5,000	24.79	12.20	84	16	8.5
6,000	23.86	11.73	81	19	10.1
7,000	22.97	11.30	78	22	11.6
8,000	22.11	10.87	76	24	13.1
9,000	21.29	10.46	73	27	14.6
10,000	20.49	10.07	70	30	16.1
11,000	19.72	9.70	68	32	17.6
12,000	18.98	9.34	65	35	19.1
13,000	18.27	8.98	63	37	20.6
14,000	17.59	8.65	60	40	22.1
15,000	16.93	8.32	58	42	23.5

efficiency standpoint if the capacity of the compressor at sea level is taken as 100 per cent. Relative values of this volumetric efficiency, the loss of capacity, and decrease in the power required for compression are given in Table VI for various elevations ranging from sea level up to 15,000 feet.

TABLE VII

COMPRESSION AT DIFFERENT ELEVATIONS

Compression at Sea Level (Normal Conditions, Temperature 60° F.)

Gauge Pressure Pounds per Square Inch	Absolute Pressure Pounds per Square Inch	Temperature Degrees Fahrenheit	Percentage of Volume of Compressed Air	
			Adiabatic	Isothermal
0	14.7	60.0	100.00	100.00
10	24.7	144.3	69.18	59.51
20	34.7	206.8	54.34	42.36
30	44.7	257.5	45.40	32.89
40	54.7	300.7	39.34	26.87
50	64.7	338.6	34.92	22.72
60	74.7	372.6	31.53	19.68
70	84.7	403.4	28.84	17.36
80	94.7	432.0	26.64	15.52
90	104.7	458.1	24.81	14.04
100	114.7	482.7	23.25	12.82
Compression at 5,000 Feet Above Sea Level (Normal Conditions, Temperature 60° F.)				
0	12.27	60.0	100.00	100.00
10	22.27	158.0	65.49	55.10
20	32.27	228.0	50.33	38.02
30	42.27	284.0	41.55	29.03
40	52.27	331.1	35.74	23.47
50	62.27	372.3	31.56	19.70
60	72.27	409.0	28.39	16.98
70	82.27	442.2	25.90	14.91
80	92.27	472.7	23.87	13.30
90	102.27	500.9	22.19	12.00
100	112.27	527.2	20.77	10.93
Compression at 10,000 Feet Above Sea Level (Normal Conditions, Temperature 60° F.)				
0	10.25	60.0	100.00	100.00
10	20.25	173.3	61.67	50.62
20	30.25	251.3	46.38	33.88
30	40.25	312.7	37.86	25.47
40	50.25	364.0	32.34	20.40
50	60.25	408.5	28.43	17.01
60	70.25	448.0	25.50	14.59
70	80.25	483.6	23.20	12.77
80	90.25	516.3	21.34	11.36
90	100.25	546.5	19.81	10.22
100	110.25	574.6	18.51	9.30

TABLE VII—(*Continued*)

Compression at 15,000 Feet Above Sea Level (Normal Conditions, Temperature 60° F.)

Gauge Pressure Pounds per Square Inch	Absolute Pressure Pounds per Square Inch	Temper-ature Degrees Fahrenheit	Percentage of Volume of Compressed Air	
			Adiabatic	Isothermal
0	8.56	60.0	100.00	100.00
10	18.56	190.6	57.72	46.12
20	28.56	277.1	42.51	29.97
30	38.56	344.0	34.35	22.20
40	48.56	399.6	29.16	17.63
50	58.56	447.5	25.53	14.62
60	68.56	489.9	22.84	12.49
70	78.56	528.1	20.72	10.90
80	88.56	563.0	19.03	9.67
90	98.56	595.2	17.64	8.69
100	108.56	625.2	16.47	7.89

It will be seen that the decrease in capacity at higher altitudes is greater than the decrease in power used. Thus, at an altitude of 3,000 feet the loss in capacity of the compressor is 10 per cent., but the power used is only 5.2 per cent. less. Therefore, for a given amount of work, not only would a larger compressor be required, but its consumption of power would be greater than for the same machine at sea level.

In Table VII are shown data in regard to air compression at different altitudes.

METHODS OF DRIVING AIR COMPRESSORS

26. Possibly 80 per cent. of the air compressors in use in all industries are belt-driven, and for industries other than mining the most common power units are electric motors. Steam engines are more common in the mining industry. In Fig. 9 is shown a compact belt-driven unit, consisting of a duplex compressor and an electric motor mounted on one foundation. The disadvantage common to the compact construction, such as heating of bearings and excessive friction losses, are overcome by the use of a loose belt and an idler pulley.

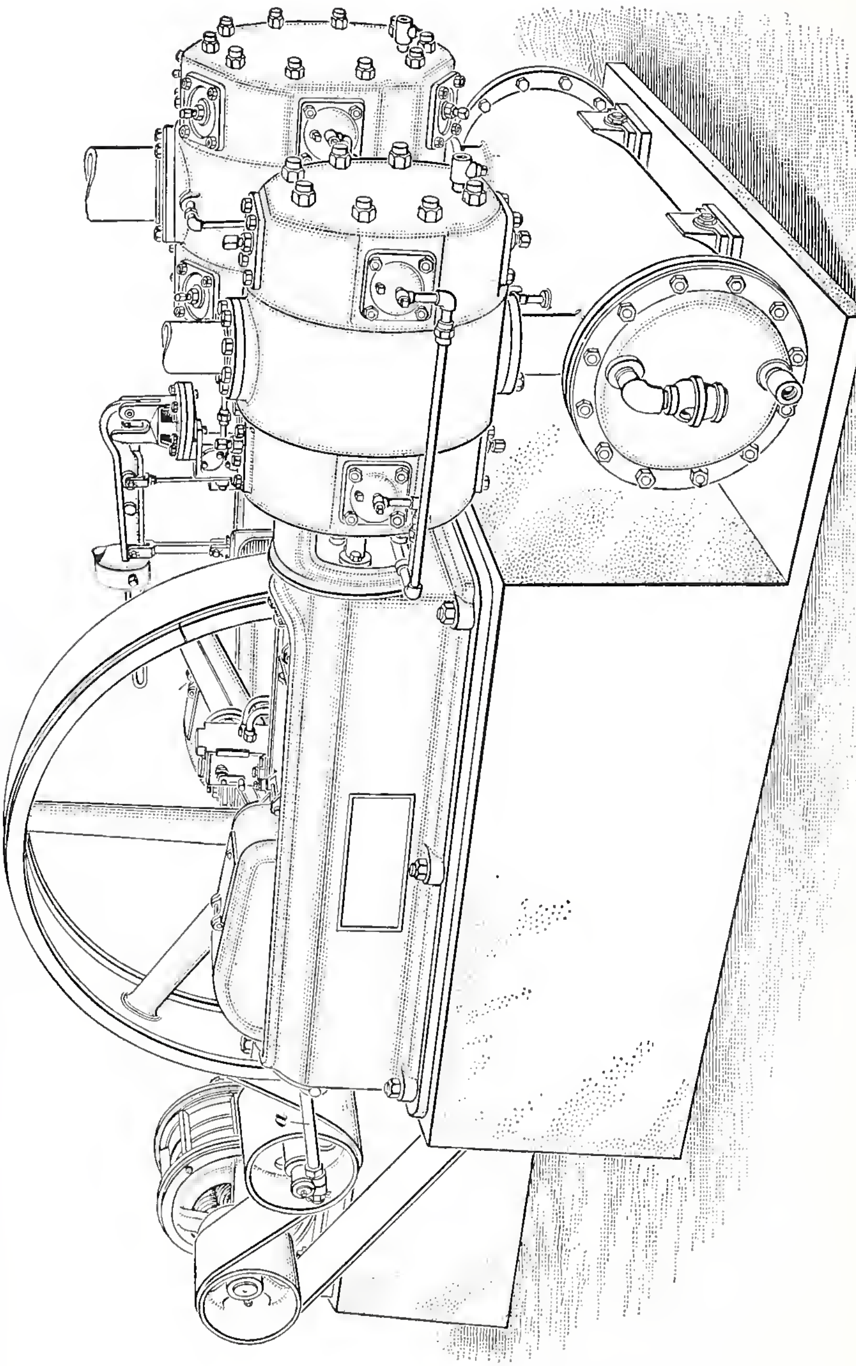


Fig. 9

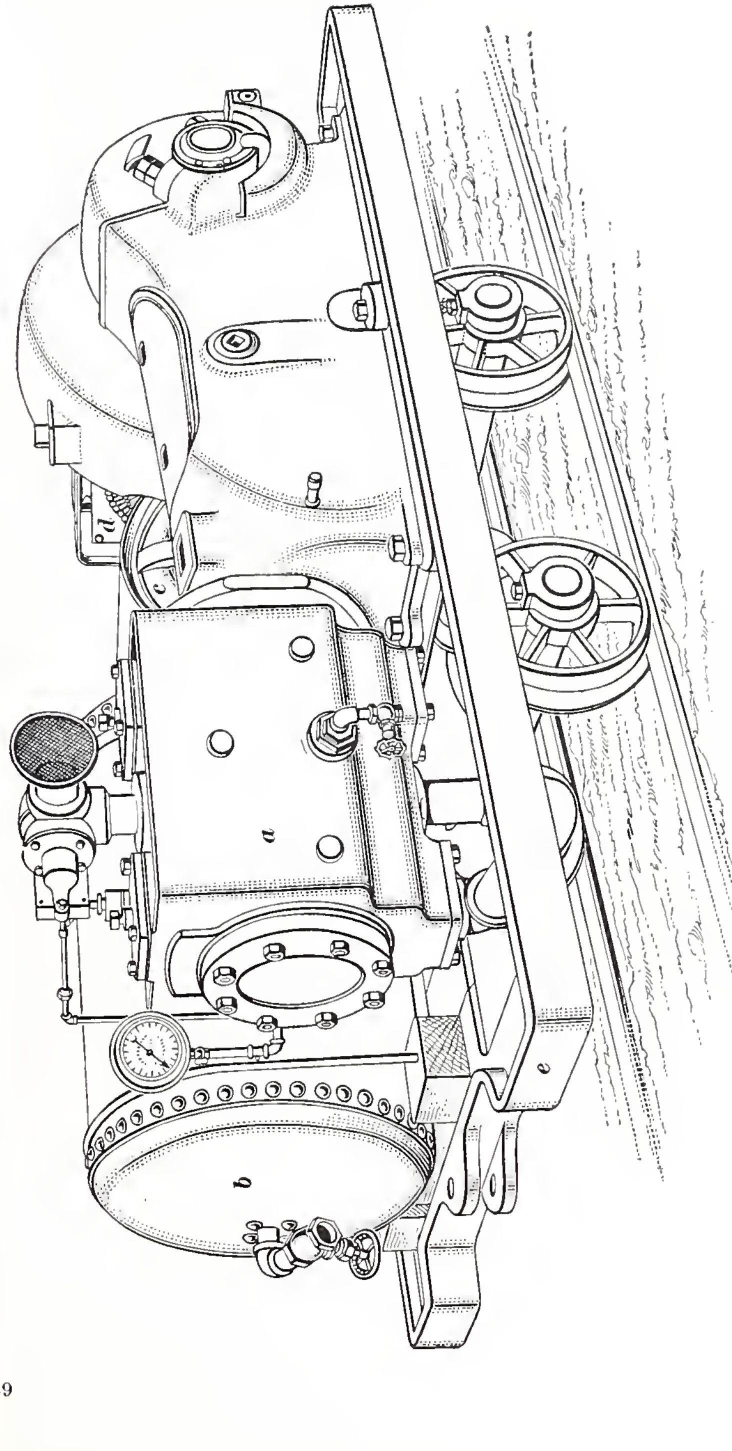


Fig. 10

The extra length of the belt gives a large arc of contact between the belt and the pulleys with the corresponding increase in power. The weight of the idler is sufficient to keep the belt from flapping and to maintain the required arcs of contact. This pulley is carried between two arms, one of which appears at *a*, hinged to the compressor frame so that the weight of the pulley rests on the belt. When the load increases, the lower side, or tight side, of the belt is drawn still tighter, the upper side slackens a little, and the idler pulley drops lower, thus increasing the arcs of contact on both the driving and the driven

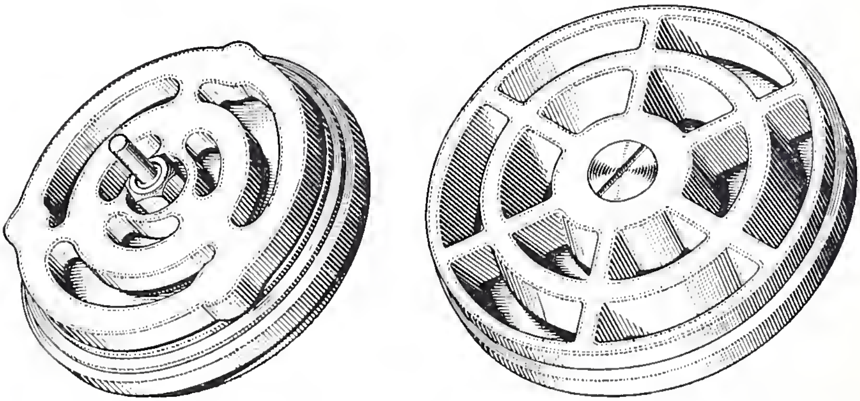


Fig. 11

pulleys. In no case should the belt be tighter than is necessary to prevent slipping; overtight belts cause heated bearings and excess losses.

For belt drives from line shafts or from units other than electric motors, long belts are generally needed; these require much floor space and are not easy to keep properly adjusted. High-speed compressors are sometimes driven by direct-connected electric motors.

27. In Fig. 10 is shown a portable, electrically-driven, single-stage compressor unit for use in non-gaseous mines that are wired for electricity. The compressor *a*, a receiving tank *b*, a motor *c*, a controller *d*, and connecting gearing, cranks, and piston rods are compactly mounted on a truck frame, so that the unit can be placed near where the work is to be done. Such a set is useful wherever compressed air is needed in comparatively

small quantities. For outside work, the power unit may be a steam or gasoline engine.

AIR-COMPRESSOR VALVES

28. Automatic Plate-Type Valves.—Automatic valves of the plate type are used for both intake and the discharge on practically all air compressors. All these valves are essentially the same in principle and they may be explained by reference to

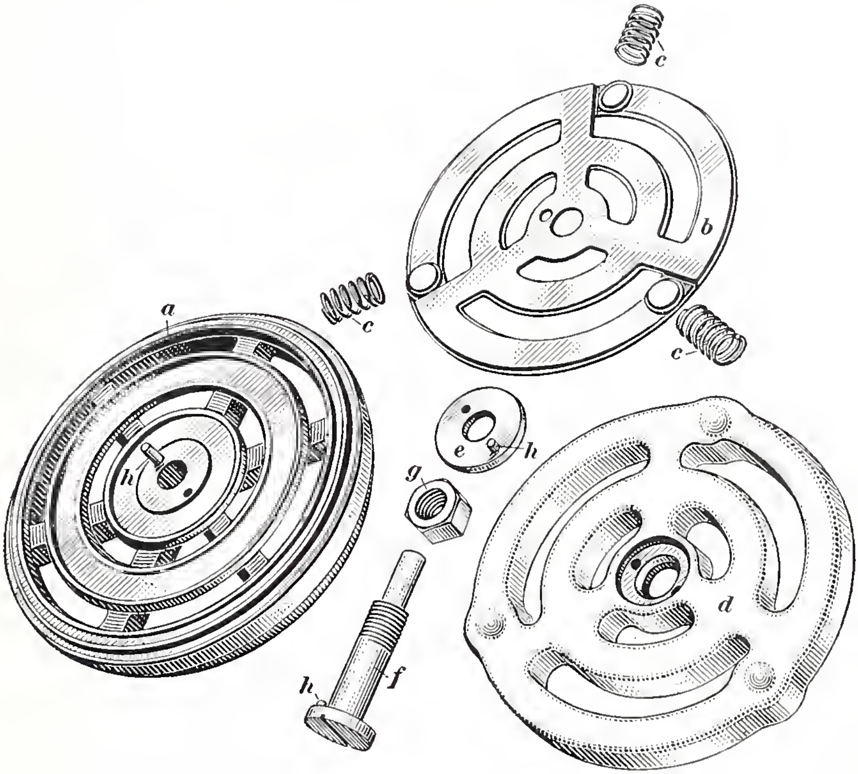


Fig. 12

Fig. 11, which shows two views of such a valve, and Fig. 12 which shows the separate parts. The valve seat *a*, Fig. 12, has annular ring openings, and when the parts are assembled these openings are covered by a valve plate *b*. The plate is held against the openings by compression springs *c* that fit over bosses on the valve plate and in depressions in the guide plate *d*. A washer *e* on the center of the valve seat acts as a spacer between the valve seat and the guide plate, and after the springs

are in place the guide plate is pressed down over the assembly and clamped by means of a bolt *f* and a nut *g*, as shown in Fig. 11. The dowel pins *h*, Fig. 12, on the valve seat, the washer, and the bolt head, prevent any relative turning of the parts.

When the air pressure tending to lift the valve plate off its seat is great enough to overcome both the opposing pressure of the air and the springs, the plate rises, allowing air to pass until the two pressures are equalized, or nearly so, when the plate reseats automatically. This type of valve is so placed in the compressor that it can be easily taken out by removing a valve cover.

29. Mechanically Operated Valves.—Valves operated mechanically by means of connections with some other moving parts of the compressor are still found. Valves similar to the Corliss-type steam valve are in some use, as are also combined rotary and poppet valves.

REGULATION OF AIR COMPRESSORS

30. Regulation of Output and Pressure.—The output of air compressors can be regulated by adjusting the intake or discharge valves, the speed, the clearance, or a by-pass through which air may pass from the discharge to the intake. For example, the intake valves can be adjusted to take air during part or all of the stroke, as required; the discharge valve can be adjusted to remain open during part of the suction stroke; a safety valve can be made to release some of the compressed air; a governor can be arranged to throttle the speed when the pressure is too high, and so on.

Automatic unloaders are in use to limit the compression automatically by reducing the intake and opening the discharge to the atmosphere at a predetermined pressure. For example, a duplex compressor may be fitted with a combination unloader, which will stop the compression entirely when the pressure has risen to the required limit and start it again when the pressure falls below a predetermined minimum. The usual range

between maximum and minimum pressures is 10 or 15 pounds, but it can be made of almost any range desired.

31. Cylinder Clearance.—To prevent damage, the piston must stop in its stroke before it strikes either end of the cylinder, thus leaving a space at each end of the cylinder over which the piston does not pass. This space at one end of the cylinder, plus all the space in the air ducts up to the valve that closes the cylinder, is called the *clearance space,* or briefly, *clearance.* At the end of the stroke, the clearance is filled with air at the delivery pressure and temperature, and at the beginning of the return stroke this air expands, its pressure falling to that of the incoming air. The compressed air remaining in the cylinder assists in overcoming the inertia of the piston at the beginning and end of each stroke, and as a result, clearance in any air cylinder is not a complete loss, although it decreases the capacity of the cylinder.

LUBRICATION OF AIR COMPRESSORS

CHOICE AND APPLICATION OF LUBRICANTS

32. The lubrication of air compressors may be considered as *external* and *internal.* Lubrication of parts of the machine that do not come into contact with the air in the process of compression is external, and lubrication of parts that do come in contact with the air in the process of compression is internal. In both cases successful lubrication demands that a film of oil be maintained between surfaces where friction would otherwise occur. The maintenance of this film on external surfaces is precisely the same problem as with the engines and motors by which compressors are driven.

Internal lubrication is best effected by means of mechanical force-feed systems, which have been developed to a point where they require little attention after being set to feed properly, except to keep them supplied with oil. The oil used should be of the very best quality. The large oil companies have worked out the exact requirements for different oils and machines. Their recommendations, together with those of the machine manufacturers, should be followed. Otherwise, the result will

be purely a matter of guesswork on the part of the man in charge. New air compressors should be supplied with two or three times the normal amount of oil for several days, until the working parts are smoothed down. The discharge valves should be examined from time to time, and if they present a dry, dusty appearance, a lack of oil is indicated.

33. The heat to which oil is subjected in a compression cylinder, though somewhat less than the actual heat of compression because of the effect of the water-jacket or other cooling arrangements on the frictional surfaces, is still high enough to cause many grades and kinds of oil to volatilize. Any oil containing ingredients that will volatilize at a point lower than the compressor temperature shown in Table VII, is likely to vaporize; and when vaporized oil is mixed with air the mixture may ignite and burn, or even explode with great force. Oils with ingredients that oxidize not only cause fire hazard, but they also reduce efficiency by forming carbon deposits that obstruct air passages and make valves leak. Leaky discharge valves cause overheating by permitting the *wire-drawing* of compressed air back through the valves into the cylinders. Frequent inspection and cleaning are necessary to overcome this tendency.

As a rule, light-body oils give free running lubrication, but their point of vaporization is lower than the general run of heavy-body oils. Heavy oils, on the other hand, are more likely to make deposits of carbon and cause the formation of sticky masses about ports and valves. The bad effect of these deposits is aggravated by the presence of dust particles in the air.

As the *quantity* and *quality* of the lubricant used in an air compressor are of great importance, the manufacturer of a compressor should be consulted as to the most suitable lubricant and the quantity. When this advice is asked, the manufacturer should be informed as to the precise conditions under which the machine is to be used, for these conditions may affect the decision. The makers of all standard types of compressors will give specific advice in such cases and will also give good general advice concerning the lubrication of all compressors.

REMOVAL OF DEPOSITS CAUSED BY LUBRICANTS

34. In every compressed-air system, whatever the quality of lubricant used, there are accumulations of carbon deposits, and these should be removed at regular intervals. The best cleansing solution is a suds made of one part soft soap to fifteen parts water. These suds should be fed into the air cylinders while the compressor is running for a few hours about once each week at a rate about ten times the normal rate of feeding oil. The suds can be fed by means of a hand pump or through

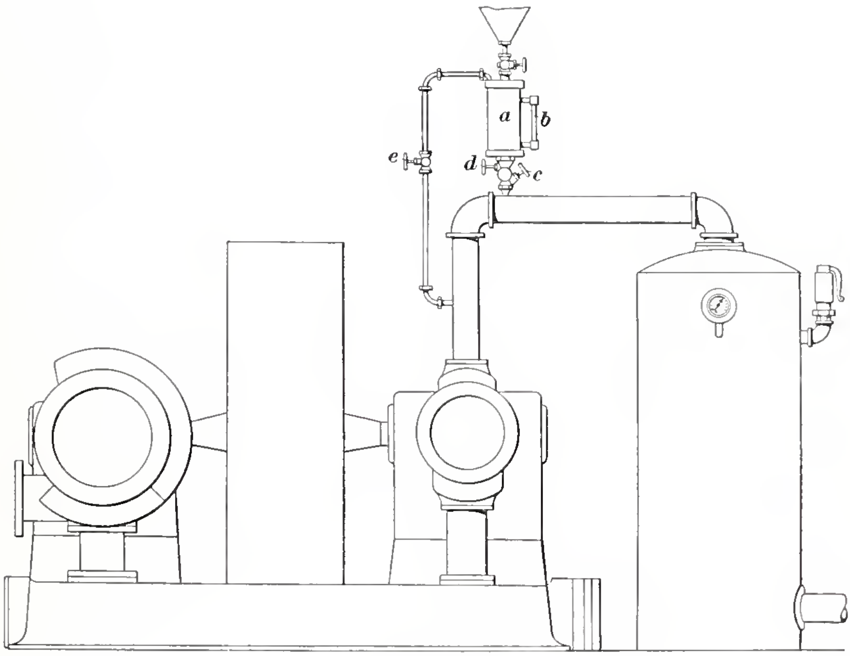


Fig. 13

the regular lubricator. The quantity needed can be ascertained by inspecting the valves and observing the deposits on them; this should be done regularly. After the suds have been used, the accumulated liquid in the air receiver and the cooler should be drawn off through the drain cocks, and before the machine is shut down it should be run again for about half an hour with the regular oil feed, so that all inner surfaces may be covered with a film of oil to prevent rusting. Kerosene, gasoline,

or other light oils should never be used in an air cylinder because of their volatile nature in the presence of heat.

35. Deposits of oil, carbon, and other refuse in air-discharge lines and receivers can be softened and loosened by means of a solution made by dissolving 1 pound of Red Seal Lye in 18 pounds of water. An arrangement for feeding this solution into the discharge line is shown in Fig. 13. A chamber *a*, made of a piece of large pipe, is fitted with a glass indicating gauge *b*. Above the chamber is a filling valve and a funnel, and below is a connection with the discharge line through a pressure cock *c* and a regulating valve *d*. The top of the chamber is also connected with the discharge line through a pressure cock *e* so that the pressure above and below the solution in the chamber can be equalized. The regulating valve should be adjusted to feed 60 or 70 drops a minute while the compressor is running, and the feeding of the lye solution should be continued until the accumulation of sediment is loosened so that it will come out of the blow-off valve on the receiver. This operation should be repeated every month or two, as required.

CARE OF AIR COMPRESSORS AND AUXILIARIES

36. Installing a Compressor.—An air compressor, like any other machine, should rest on a firm foundation large enough and heavy enough to resist vibration. Compressor manufacturers generally furnish templates or drawings for constructing foundations.

Precautions should be taken to get intake air as free as possible from dust. The intake opening should usually be outside the compressor house; it should be elevated possibly 8 or 10 feet above the ground, and should be well screened.

37. Air Receivers.—The receiver into which the compressor discharges should be of ample size to absorb the pulsations of air so that the flow of air into the pipe line will be steady and uniform. Large receiver capacity is an advantage, because it helps to steady the supply and to compensate in some

degree for the irregular use of air when a number of machines are supplied by one compressor.

Vertical receivers are generally preferred. The main receiver should not be far from the compressor or the after-cooler. The receiver may, however, be placed outside the building, where further cooling effect and condensation of moisture may occur in it. If the receiver is exposed to the weather it should be well painted, especially around the joints and all places where rusting with resulting leakage is most likely to occur. There may be one or more secondary receivers along the pipe line to help control pulsations, cool the air, and collect water of condensation. The main receiver near the compressor should have a safety valve and this valve should be inside the compressor house where the temperature is always high enough to prevent freezing. Each receiver should have a drain cock at its lowest point, and accumulated water should be drawn off at frequent intervals. Water should not be allowed to freeze in a receiver.

38. Piping.—The discharge pipe from a compressor should be short, it should be amply large, the necessary changes of direction should be long easy bends instead of elbows, and if the receiver is vertical the discharge pipe should enter the top of it. *A discharge pipe should contain no shut-off valve between the compressor and the receiver unless between this valve and the compressor is a safety valve.* Independent pipes should be installed for carrying cooling water to the jackets of the compressor cylinders, to the inter-cooler, and to the after-cooler. Pipes to carry steam to and from the engine driving a compressor are installed according to the rules for steam-engine work.

Pipe lines from the receiver to the places where the air is to be used should also be amply large and as free as possible from elbows and sharp bends. With such piping the transmission losses are minimized. Joints should be made tight with red lead. Low spots where water may remain should be avoided and provisions made so that all water will run to traps from which it can be drained occasionally. In the installing of a pipe line, provisions should be made for outlets with proper

valves, at points where they are likely to be needed in the future.

If heavy pipe is needed for high-pressure air, a larger size may be necessary in order that the thicker walls may not restrict the inner diameter too much.

39. Packing.—High-pressure fibrous packing made largely of asbestos mixed with a lubricant that includes graphite is best for the stuffingboxes of air compressors. This packing comes in strips, and a stuffingbox is filled with rings of the packing so placed in the box that the joints of the several rings are staggered. When the box is full, the gland should be screwed up by hand enough to put some pressure on the packing. Too much pressure is injurious because it squeezes out the lubricant and tends to make the packing hard. After the compressor is warmed up, the gland should be tightened *only enough to prevent leakage of air.* Too much tightening of a gland at any time is likely to cause heating and scoring or cutting of the piston rod. Hardened packing can sometimes be renewed by loosening it up and working in with it a mixture of graphite grease.

Gaskets for air cylinder heads may be made of any good asbestos sheet packing, but in renewing a gasket care must be taken to use a sheet of the original thickness in order to preserve the clearance. Rubber sheet packing is not satisfactory, because it is softened by oil and heat.

AIR-COMPRESSOR TERMS AND CALCULATIONS

40. Various Terms Used.—Among terms used in discussing air compressors are *displacement, capacity, volumetric efficiency, compression efficiency, mechanical efficiency,* and *over-all efficiency.* Their meanings and the calculations required in connection with them will now be explained.

41. Displacement.—When the piston of an air compressor moves through a compression stroke, it displaces a volume of air equal to the product of the cross-sectional area of the piston—that is, the *active area*—and the distance through which it moves. This volume is the displacement per stroke, and when multiplied

by the number of strokes per minute the product is the displacement per minute, which is usually expressed in cubic feet. In a single-acting cylinder, the number of compression strokes per minute is the same as the number of revolutions per minute; in a double-acting cylinder, the number of compression strokes is twice the number of revolutions per minute. In a single-acting cylinder the entire area of one side of the piston is in contact with compressed air and is the active areas, while in a double-acting cylinder the areas of both sides of the piston, less the cross-sectional area of the shaft, are active, and the net area is the average of the two areas. The formula for calculating piston displacement is

$$D = A \times L \times N$$

in which D=displacement in cubic feet per minute;
A=active piston area, in square feet;
L=length of stroke, in feet;
N=number of active strokes per minute.

Example 1. A single-acting single-stage compressor with a 12-inch piston and a 10-inch stroke runs 235 revolutions per minute. What is its displacement in cubic feet per minute?

Solution.—The active piston area in square feet is $A=\frac{12^2\times.7854}{144}=.7854$; the length of the stroke in feet is $L=\frac{10}{12}=\frac{5}{6}$; the number of strokes per minute is $N=235$, and the displacement is $D=.7854\times\frac{5}{6}\times 235=154$ cu. ft. nearly. Ans.

Example 2.—If the compressor referred to in example 1 is double-acting and the piston rod is 1 inch in diameter, what is the displacement?

Solution.—The active areas of the two sides of the piston is $2\times.7854$ less the cross-sectional area of the piston rod, or $2\times.7854-\frac{1^2\times.7854}{144}=1.5708-.00546=1.5653$ sq. ft. The net, or average, area is therefore $\frac{1.5653}{2}=.7826$ sq. ft. The length of stroke $L=\frac{5}{6}$ ft., as before, and the number of strokes $N=2\times 235=470$. Then, $D=.7826\times\frac{5}{6}\times 470=306+$cu. ft. Ans.

42. Capacity of a compressor, generally expressed in cubic feet per minute, is the quantity of free air actually drawn into the compressor and compressed. The capacity rating of a com-

pressor is based on an intake temperature of 60° F. and atmospheric pressure of 14.7 pounds per square inch. If these conditions change, the capacity of the compressor will also change; so the exact capacity of a compressor can only be determined by actually measuring the volume of the air delivered. Information about the special appliances needed for such measurements can be obtained from compressor manufacturers.

43. Volumetric Efficiency.—The volumetric efficiency of a compressor is the ratio of its capacity to its displacement. This efficiency depends on the ratio of the clearance to the displacement. The larger the clearance space, the lower is the volumetric efficiency, because compressed air in this space must expand below atmospheric pressure before free air can enter back of the retreating piston. The product of the volumetric efficiency and the displacement is the theoretical capacity. Conversely, the capacity divided by the volumetric efficiency gives the displacement. For example, if a capacity of 850 cubic feet of free air is needed and a manufacturer guarantees his compressors to have a volumetric efficiency of 85 per cent., the compressor selected must have a displacement of $850 \div .85 = 1{,}000$ cubic feet.

44. Compression Efficiency.—The *compression efficiency* is the ratio between the amount of work theoretically required to compress the air adiabatically and the work actually required as determined from an indicator diagram. The same working conditions and air delivery are considered in both cases.

45. Mechanical Efficiency.—The *mechanical efficiency* is the ratio of the power output as represented by the energy absorbed in compressing the air and the power input required to drive the compressor. The loss of efficiency is due to overcoming frictional resistances in the compressor, and will vary from 90 to 95 per cent. This ratio is best determined by taking indicator diagrams from the air cylinder and the steam cylinder at the same time and computing the power represented by each. If the compressor is power-driven the brake horsepower of the

driving mechanism is used instead of the indicated horsepower of the steam cylinder.

46. Over-All Efficiency.—The *over-all efficiency* is the ratio that the power that would be required to compress the air if there were no losses bears to the power actually required. This ratio is the product of the compression efficiency times the mechanical efficiency. If it were not for friction losses in the compressor and those losses due to the heat of compression, the work represented by the air delivered would be equal to the work delivered to the compressor. This ratio shows the proportion of the power delivered to the compressor that is used to advantage and the proportion that is non-productive.